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FEASIBILITY STUDY OF A TUNGSTEN WATER-MODERATED NUCLEAR ROCKET

VI. FEED SYSTEM AND ROTATING MACHINERY

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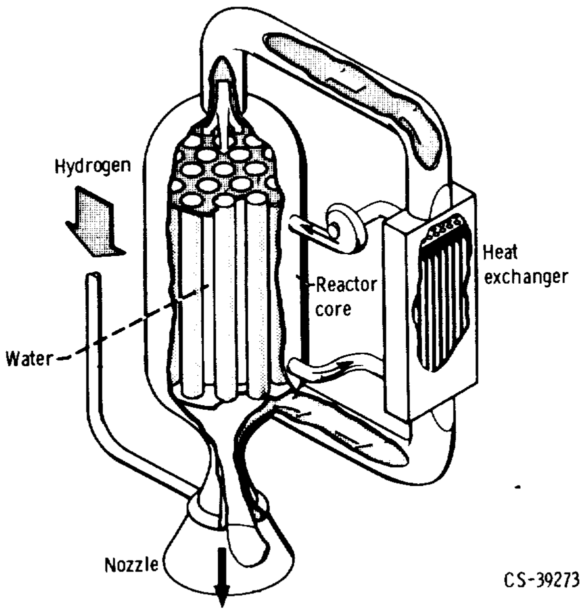
PREFACE

The concept of a nuclear rocket system based on the use of a tungsten water-moderated reactor (TWMR) was originated at the Lewis Research Center. The TWMR is a thermal reactor that uses water as the moderator, uranium dioxide as the fuel, and tungsten enriched in tungsten 184 as the fuel element structural material. As is common to all nuclear rocket systems, hydrogen is used as the propellant to maximize specific impulse. The reactor (see illustration) consists of a tank containing a number of pressure tubes that are attached to tube sheets at the inlet and outlet ends of the reactor. The pressure tubes contain the fuel elements. The space inside the tank between the tubes is filled with water, which serves both as the neutron moderator and as a coolant for the structure. Heat is generated in the water by neutrons and gamma rays and is also transferred to the water by heat leakage from the hot fuel elements, each of which is located in a pressure tube. The removal of heat is provided by pumping the water through the core and a heat exchanger in a closed loop. The water is regeneratively cooled in the heat exchanger by the hydrogen propellant, which flows from a supply tank through the nozzle and heat exchanger into the core. As the hydrogen flows through the core pressure tubes and through the fuel elements, it is heated to a high temperature and is expanded out the nozzle to produce thrust.



The potential advantages of the concept lie in the following areas: The use of tungsten provides a high-temperature material with good thermal shock resistance, tensile and compressive strength, thermal conductivity, and resistance to corrosion by the hydrogen propellant. The properties of tungsten also permit the fabrication of fuel elements with very thin cross sections for good heat transfer. The use of water as the moderator provides a good coolant for the pressure vessel and structural members and reduces core size and weight over that obtained for most moderator materials. In this concept, the fuel element assemblies are structurally independent of each other and thus permit individual development of these assemblies.

A program was undertaken at Lewis to investigate the engineering feasibility and performance of the TWMR nuclear rocket system. The results of these investigations, which are summarized in part I (NASA Technical Memorandum X-1420) of this series of reports, are presented in detail in the other six parts of the series as follows: II. Fueled Materials (NASA Technical Memorandum X-1421); III. Fuel Elements (NASA Technical Memorandum X-1422); IV. Neutronics (NASA Technical Memorandum X-1423); V. Engine System (NASA Technical Memorandum X-1424); VI. Feed System and Rotating Machinery (NASA Technical Memorandum X-1425); VII. System Dynamics (NASA Technical Memorandum X-1426).

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APPENDIXES

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FEASIBILITY STUDY OF A TUNGSTEN WATER-MODERATED NUCLEAR ROCKET

VI. FEED SYSTEM AND ROTATING MACHINERY (U)

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SUMMARY

The study of the tungsten water-moderated reactor (TWMR) feed system and rotating machinery included a theoretical investigation of feed system types; the design of a specific reference feed system; the selection of pumping machinery for the reference feed system, moderator system, and poison control system; and the arrangement of components of these systems which are external to the reactor vessel.

Two basic feed system types were first considered, a hot-bleed and a topping system. The study of the hot-bleed system indicated that a bleed rate in excess of 5 percent of pump flow would be required. The high bleed rate was considered undesirable as it reduces the specific impulse of the rocket. The chief drawback of the topping system for the TWMR engine was its lack of uprating potential (i.e., a future increase in engine performance by increasing the propulsion nozzle flow rate and chamber pressure). The reference design chamber pressure of 600 psia (414 N/cm^2 abs) is near the upper limit of topping system feasibility due to the low energy gas available to drive the turbine.

By combining the bleed and topping systems, the objectionable features of each can be avoided. This combination, called the split system, was selected as the reference feed system for the TWMR engine study. The reference feed system contains two hydrogen pumps in series, one driven by a bleed turbine and the other by a topping turbine. The bleed rate required by the reference feed system was 2.6 percent, and the resulting engine specific impulse was 829 seconds.

The TWMR reference engine has two other flow systems, the moderator cooling system and the chemical poison control system. Each has a circulating pump driven by a bleed turbine. An arrangement drawing was made of the reference engine showing all major components associated with the three fluid systems which are external to the reactor vessel.

The requirements of all four engine turbopumps were studied, and mechanical designs were carried far enough to be reasonably sure such units could be developed. Predicted performance curves, and estimates of weight, fluid holdup volume, wetted surface area, and moment of inertia of rotating parts were made for each unit. No major problem areas were revealed during the turbopump study.

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INTRODUCTION

In the nuclear rocket concept presented, the hydrogen, stored at low temperature and pressure, must be pumped through the reactor and delivered to the propulsion nozzle at high temperature and pressure so that it can be expanded to produce thrust. This is the function of the feed system. As used herein, the engine feed system includes the pumps, drive turbines, valves, piping, heat exchangers, and reactor components through which the hydrogen flows on its way from the storage tank to the propulsion nozzle chamber. It also includes bleed system components in systems where a portion of the main hydrogen flow is bled off to drive one or more turbines.

The TWMR engine also contains two auxiliary fluid systems. One is the moderator system in which the water moderator is circulated between the reactor core and a water-to-hydrogen heat exchanger for cooling purposes, and the other is the liquid poison control system in which a water solution containing variable amounts of a neutron absorber is recirculated to control reactor power. Both of these systems contain circulating pumps, valves, piping, and heat exchangers. The turbines for the circulating pumps are driven by feed system hydrogen.

In order to study the rocket concept in depth, a specific reference-design engine was chosen. Selection and design of a suitable feed system for the reference engine was the primary purpose of this study. The design of feed system components such as turbopumps was only to be carried far enough to assure feasibility. The work included a study of feed system cycles, the selection of turbomachinery for all three fluid systems, the prediction of turbomachinery performance characteristics, the arrangement of engine components external to the reactor vessel, and the sizing of valves and piping for the hydrogen and water systems.

The study was analytical in nature and the design approach was conservative. Because most of the engine components were closely interrelated and were being designed at the same time, the study involved a number of equipment changes and the numerical iterations. At one point, a basic change was made in the reference-design requirements which caused the reactor power rating to be reduced from 1725 megawatts to the final 1500 megawatts. While some of the material presented in this report was based on the earlier requirements, the conclusions still apply.

At the beginning of the study, certain operating conditions were specified. These included reactor power, propulsion nozzle chamber temperature and pressure, and hydrogen flow rate (see table I). A general philosophy regarding design criteria was developed early in the program to serve as a guideline. The major points were as follows:

(1) The design of all components was to be conservative, that is, well within the current state of the art.

(2) Component ratings were to be flexible to accommodate minor changes without redesign.

(3) The engine feed system type was to have a potential uprating capability, that is, be feasible at higher chamber pressures, power levels, and flow rates.

The general procedure followed during the program is outlined as follows:

(1) Study feed system cycle types and select one for the reference engine.

(2) Study turbomachinery types and make selections for all fluid systems of the reference engine.

(3) Make tentative equipment arrangement layout.

(4) Size valves and piping and accurately calculate system pressure drops and temperature changes for all systems.

(5) Make mechanical design layouts of all turbomachinery.

(6) Develop turbomachinery performance curves.

(7) Prepare information for transient studies (startup, shutdown, control, and overspeed).

(8) Check valve and piping size for reduced power operation; make necessary changes to equipment arrangement layout, pressure drop calculations, and turbomachinery operating points.

(9) Compare final reference feed system with other types of feed systems.

The arrangement of this report differs somewhat from the preceding study procedure. The first section covers the initial study of several feed system cycles. The second section presents details of the final reference feed system including flow schematics, temperature and pressure data, and equipment arrangement. The reference feed system is then compared with two other types, all based on the reference reactor. The third section covers the feed system turbomachinery. A number of factors considered in selecting this machinery are discussed, and details of the reference-design machinery are presented. The next two sections deal with turbomachinery for the moderator and poison systems, respectively. Each system is described, and details of the reference-design machinery for these systems are presented. The last section contains a discussion of the results and the conclusions reached. Background data used in the reference-design calculations are included in the appendixes.

Members of the Lewis staff who contributed substantially to the study program by assisting the author in their specialized areas include Melvin J. Hartmann, George Kovich, Robert E. Connelly, Walter M. Osborn, Warren J. Whitney, and Walter A. Paulson.

SYMBOLS

a exponent of mass flow rate ratio defined in eq. (A1)

b exponent of density ratio defined in eq. (A1)

C_V	flow coefficient defined in appendix C
C_x	constant defined in eq. (B1)
C_y	constant defined in eq. (B4)
C_z	constant defined in eq. (B4)
c	exponent of absolute pressure ratio defined in eq. (A1)
c_p	specific heat at constant pressure, Btu/(lb mass)(^{o}R); J/(kg)(^{o}K)
DN	bearing parameter, bearing bore diameter times speed, mm-rpm
d	exponent of temperature ratio defined in eq. (A1)
FPE	fluid property effect, ft; m
H_g	head due to gravity, ft; m
H_L	head loss, ft; m
H_{sv}	net positive suction head, ft; m
H_v	fluid vapor head, ft; m
Δh	isentropic enthalpy change, Btu/lb mass; J/kg
N	rotational speed, rpm
NPSP	net positive suction pressure, psi; N/cm^2
n	effective number of turbine stages, dimensionless
P	absolute pressure, lb force/ft^2 abs; N/cm^2 abs
ΔP	pressure differential, lb/ft^2; N/cm^2
p	absolute pressure, psia; N/cm^2 abs
Δp	pressure differential, psi; N/cm^2
Q	flow rate, gal/min; m^3/min
SL	system pressure loss, psi; N/cm^2
S_s	suction specific speed, $N(Q)^{1/2}/(H_{sv} + FPE)^{3/4}$
T	temperature, ^{o}R; ^{o}K
$\dot{w}$	mass flow rate, lb mass/sec; kg/sec
γ	ratio of specific heats, dimensionless
η	efficiency, percent
ρ	density, lb mass/ft^3; kg/m^3
τ	torque, lb force-ft; J

Subscripts:

max	maximum
o	overall
p	pump
s	stage
t	turbine
1	condition 1, inlet, or station at pump inlet (fig. 2)
2	condition 2, outlet, or station at pump discharge (fig. 2)
3	station at topping turbine inlet (fig. 2)
4	station at topping turbine outlet (fig. 2)
5	station at propulsion nozzle chamber (fig. 2)

FEED SYSTEM CYCLE STUDIES

The first task of the program was to study feed system cycles and to select one suitable for the TWMR reference engine. This section is concerned with the background study that was made of several cycle types and upon which the selection was based. The studies presented in this section were based on the initial specified operating conditions given in table I, for a 1725-megawatt reactor system. While the specified conditions were later changed, the cycle characteristics, limitations, and conclusions reached still apply. Calculations indicated that the temperature of the hydrogen gas leaving the water-to-hydrogen heat exchanger would be in the general vicinity of 320° R (178° K), and that the system pressure loss SL would be approximately 450 psi (310 N/cm^2) for the 1725-megawatt system. The system pressure loss is defined herein to consist of all the pressure drops in the feed system from the hydrogen storage tank to the propulsion nozzle chamber, excluding the pressure changes across the pump or turbine.

Bleed System

A simplified hot-bleed system is illustrated in figure 1. Hydrogen is pumped from its storage tank through the nozzle cooling jacket, the heat exchanger, and the reactor core. After passing through the reactor core, the flow divides into two components. The major portion of the flow passes through the propulsion nozzle to produce thrust. The remaining hydrogen, the bleed flow, is used to drive the turbine and is then ejected overboard at conditions which may allow for as much as a 50-percent recovery of the specific impulse of that portion of the flow.

At the initial conditions specified in table I, a three-stage turbine may attain about 53 percent efficiency. If a pump efficiency of 65 percent were assumed and a 20-percent turbine flow bypass were allowed for control purposes, the system would require about 5.7 percent bleed flow. In the event that the pump experienced a 5-percent increase in efficiency, the bleed rate required would decrease to 5.3 percent. If the system pressure loss was increased from 450 to 650 psi (310 to 448 N/cm^2), or the chamber pressure was increased 200 psi (138 N/cm^2), the bleed rate required would increase from 5.7 to 6.6 percent. These values are indicative of the rather large changes in bleed flow that are encountered as a result of system or component variations.

In the concept that is presented in this report, several of the reactor fuel elements are specifically designed to produce bleed hydrogen at a temperature of 1860° R (1030 $^{\circ}$K) so that it will be compatible with present day turbine materials. Other means could be used to produce a compatible temperature such as mixing hot gas, bled from the nozzle chamber, with cold gas or liquid. This method would require a hot mixing valve which could be difficult to develop. The bleed fuel elements, which are located in the center of the core, should form a symmetrical pattern for neutronic reasons. The number of bleed elements in the TWMR core can be one, three, four, seven, etc. Any change in the bleed flow rate or the operating conditions requires close coordination of reactor and turbopump design. The number of elements to be initially committed to bleed flow should be capable of providing for increased bleed rates without major redesign of the reactor core. Since each element could contribute approximately 1 percent hydrogen bleed, seven bleed elements were chosen for the all-bleed system. In the event that the system should be derated or the requirements decreased, the high bleed flow might not be recoverable without system modification.

A more detailed treatment of the subject of bleed systems for hydrogen-nuclear rockets is given in reference 1. That report, which evolved from the cycle studies made for the TWMR project, covers the effect of chamber pressure, system loss, and turbine efficiency on bleed rate, and shows the method used to calculate bleed rate.

Topping System

A schematic drawing of a simplified topping system is shown in figure 2. In this case, the hydrogen is pumped from the storage tank through the nozzle cooling jacket and heat exchanger to the turbine. Although all the flow goes through the turbine, only a small amount of work per pound mass (kg) is extracted because of the low temperature level. In contrast, the bleed system extracts a large amount of work per pound mass (kg) from a small flow. In the topping system, a single-stage turbine operating at 320° R (178° K) can effectively extract the required power. All the flow is then passed into the reactor and thrust nozzle. The fact that all the propellant is used at high specific impulse and the fact that relatively high turbine efficiencies can be obtained in a single-stage turbine are desirable. In general, the topping system is not particularly adaptable to uprating. Maintaining power and control margins, which may be sensitive to system conditions, becomes a problem. These facts are illustrated in the following discussion.

Reference 1 presents a detailed thermodynamic analysis of a topping system for a wide range of nozzle chamber pressures, system pressure losses, and topping turbine inlet temperatures. This type of analysis was used in the TWMR background study, but to avoid repetition the discussion in this report is confined to the following narrow range of variables of particular interest to the reference design:

Nozzle chamber pressure, p_5, psia; N/cm^2 abs 800; 552
System pressure loss range, SL, psi; N/cm^2 . . . . 350 to 650; 242 to 448
Turbine inlet temperature range, T_3, °R; °K . . . . 250 to 400; 139 to 222

System pressure loss for the topping system is

$$SL = (p_2 - p_3) + (p_4 - p_5)$$

where the subscripts refer to the station numbers shown in figure 2. The loss between the storage tank and pump is considered negligible. It was assumed that five-ninths of the system loss occurred before the topping turbine and four-ninths of the loss occurred downstream from the turbine (see ref. 1). This assumption was based on a preliminary feed system design for the reference conditions.

By curve fitting, the specific pump work required can be represented in equation form as

$$\Delta h_p = 42.822\times10^{-3}\, p_2 - 1.0203\times10^{-6}\, p_2^2 - 1.28 \tag{1}$$

or in joules per kilogram,

$$\Delta h_p = 144.40\, p_2 - 4.9899\times10^{-3}\, p_2^2 - 2976$$

This equation represents the isentropic enthalpy rise per pound mass (kg) of fluid from saturated liquid at 30 psia (20.7 N/cm^2 abs) to the required pump discharge pressure. Equation (1) is based on para-hydrogen properties given in reference 2.

Pump discharge pressure is dependent on chamber pressure, system pressure loss, and topping turbine pressure ratio as follows:

$$p_2 = \frac{p_3}{p_4}\left(p_5 + \frac{4}{9}\,SL\right) + \frac{5}{9}\,SL$$

The specific isentropic work output of the turbine depends on the turbine inlet temperature and the pressure ratio. The thermodynamic tables of reference 2 were used to obtain the specific isentropic enthalpy drop Δh_t across the turbine. The turbine inlet pressure p_3 was assumed to be 1500 psia (1030 N/cm^2 abs) to simplify the Δh_t calculation.

In order to match the pump and topping turbine, their actual specific enthalpy changes were equated as follows:

$$\frac{\Delta h_p}{\eta_p} = \Delta h_t \eta_t \times 10^{-4}$$

or expressed in terms of specific isentropic pump work,

$$\Delta h_p = \Delta h_t \eta_t \eta_p \times 10^{-4} \tag{2}$$

Figure 3(a) shows the match points obtained by assuming conservative pump and turbine efficiencies of 65 and 70 percent, respectively, referred to as case A in the following discussion. The solid curves are a plot of equation (1) and represent work required. The dashed curves came from equation (2) and represent work available. The topping turbine pressure ratio at matched conditions is obtained from the intersection of these curves for the system loss and temperature conditions being considered. At the predicted system loss of 450 psi (310 N/cm^2) and turbine inlet temperature of 320° R (178° K) for the 1725-megawatt engine, the topping turbine pressure ratio would be 1.79, and the pump discharge pressure p_2 would be 2040 psia (1408 N/cm^2 abs).

In case B, 20 percent of the gas flow is bypassed around the turbine as a margin for possible increases in design requirements and for control use. Equation (2) becomes

$$\Delta h_p = \Delta h_t \eta_t \eta_p 0.80 \times 10^{-4} \tag{3}$$

Work matching curves based on equations (1) and (3) are shown in figure 3(b). In this case, a 320° R (178° K) work-available curve would never reach the 450 psi (310 N/cm^2) work-required curve. Thus, this combination of conditions will not produce a feasible feed system.

The effect of changing component efficiency is shown in case C, figure 3(c). This figure is the same as figure 3(b) except that the pump efficiency was increased from 65 to 76 percent and the turbine efficiency from 70 to 80 percent. The turbine pressure ratio for the predicted system loss of 450 psi (310 N/cm^2) and a gas temperature of 320° R (178° K) with these optimistic efficiencies would be 1.65. The corresponding pump discharge pressure would be 1900 psia (1310 N/cm^2 abs).

To show the effect of turbine inlet temperature on turbine pressure ratio better, match point data from figure 3 were replotted for a system loss of 450 psi (310 N/cm^2), resulting in figure 4. The system represented by curve B, for instance, would require a turbine inlet temperature above 350° R (194° K) to be feasible because there is insufficient work available from the turbine for a work match below this temperature.

Likewise, the effect of system pressure loss on turbine pressure ratio is shown in figure 5. Here, match point data from figure 3 were replotted for turbine inlet temperatures of 300° R (167° K) and also 350° R (194° K). An increase in system loss is similar in effect to a decrease in turbine inlet temperature.

The most significant points brought out in the topping system study are as follows:

(1) A topping system has a definite work limitation which is primarily a function of chamber pressure, system loss, turbine inlet temperature, and component efficiency. For TWMR type engines, which have a low turbine inlet temperature, this limit occurs at chamber pressures in the vicinity of 600 to 800 psia (414 to 552 N/cm^2 abs) - the same range that is of interest in current nuclear rocket studies.

(2) Topping systems are sensitive to any factors that change the work required by the pump or the work available from the turbine, whether it be component efficiency, turbine gas bypass, turbine inlet temperature, or system pressure loss. As the turbine approaches its maximum output, this sensitivity to any change increases. For example, the system represented by curve C (long dashes) in figure 5 can stand a sizable increase in system loss with little increase in turbine pressure ratio. The less efficient system represented by curve B (short dashes) requires a large increase in pressure ratio for even a small increase in system loss, as the turbine is near its work limit for an inlet gas temperature of 350° R (194° K). The major drawback to system sensitivity is that the turbomachinery may have to be redesigned if the expected turbine gas temperature, system loss, or component efficiencies are not met.

(3) Pressure levels between the pump and turbine in a topping system could get excessively high at the higher turbine pressure ratios. High pressures would result in a heavy propulsion nozzle, heat exchanger, turbomachinery, and piping and could present serious design problems in the nozzle and pump areas. This point is clarified in the discussion of the split system which follows.

(4) While the preceding discussion has been confined to the initial design conditions specified in table I, it can be shown (see ref. 1) that an increase in chamber pressure will increase the specific pump work required without changing the specific work available from the turbine. Thus, for a given system loss and turbine inlet temperature, a higher

turbine pressure ratio is required. The effect of increasing chamber pressure is somewhat similar to that obtained by an increase in system loss at a fixed chamber pressure.

Split System

A split system evolves when the marginal character of the topping system is avoided by doing a portion of the work in a bleed system. A schematic drawing of such a pumping system is shown in figure 6. Hydrogen flows from storage through the first- and second-stage pumps in series, through the nozzle wall and moderator heat exchanger where it picks up heat, through the turbine which drives the second-stage pump, and then into the reactor core. A small portion of the propellant is bled from specific reactor fuel elements to provide hot gas for the turbine driving the first-stage pump. The balance of the gas passes through the propulsion nozzle to produce thrust. The split cycle thus combines the characteristics of the topping and bleed systems in varying degrees. In general, the necessary power margins are provided without requiring excessive bleed rates. More machinery components are required and the system becomes more complex; however, each component becomes less critical and can be designed for greater margins. Selection of the proper work split between the topping and bleed portions of the cycle can provide for a fairly wide range of operating conditions without destroying system feasibility.

The effect of work split on the topping turbine pressure ratio is illustrated in figure 7. The ratio of the work developed by the topping turbine to the total required pump work is plotted against pressure ratio for system pressure drops of 450 and 650 psi (310 and 448 N/cm^2). An all-topping system occurs at a work ratio of 1.0, and an all-bleed system occurs at a work ratio of zero. Curves B and C represent systems with low and high component efficiencies, respectively. For small work ratios, an increase in system pressure loss of 200 psi (138 N/cm^2) requires almost no increase in work from the topping turbine in terms of pressure ratio, but rather substantial increases in bleed flow are required. At larger work ratios the portion of work required from the topping turbine begins to become significant in terms of the necessary pressure ratio.

Figure 8 is similar to figure 7 except that work ratio is plotted against second-stage pump discharge pressure instead of turbine pressure ratio. Pump discharge pressure increases rapidly as the turbine approaches its maximum output capability. This region should be avoided in order to limit pressures in the system to reasonable values. Figure 8 clearly shows the advantage that a bleed system has over a topping system in regard to pressure levels. For a system loss of 450 psi (310 N/cm^2), a bleed system would have a pump discharge pressure of 1250 psia (862 N/cm^2 abs), while an equivalent topping system would require a pump discharge pressure of at least 2000 psia (1380 N/cm^2 abs).

The system represented by curve B in figures 7 and 8 is of primary interest as it is in line with the general design philosophy which was established early in the study. As was also noted in the topping system discussion, figures 7 and 8 show that an all-topping system cannot be achieved with a turbine inlet temperature of 300° R (167° K). In applying a split-feed system to the initial 1725-megawatt design conditions, where the predicted system loss was 450 psi (310 N/cm^2) and the topping turbine inlet temperature was expected to be between 300° and 350° R (167° and 194° K), a work split of 0.67 would be a reasonable compromise. This work split would give a topping turbine pressure ratio of 1.6 and a second-stage pump discharge pressure of 1850 psia (1276 N/cm^2 abs), based on a turbine inlet temperature of 300° R (167° K). These values would decrease, of course, if higher turbine temperatures were realized. If the system loss should increase from 450 psi (310 N/cm^2) to 650 psi (448 N/cm^2), the design would still be feasible. The turbopumps could be designed to operate at either system loss merely by changing speed. Neglecting changes in component efficiency and flow coefficient effects, the speed change required would be in the vicinity of 5 to 10 percent. It is also possible to compensate for an increase in system pressure loss by increasing the speed of the topping turbopump only. While this speed increase would increase the work ratio, it would be desirable in that the bleed system components and bleed flow rate would be unaffected. The bleed rate required for the 1725-megawatt split system, based on a system loss of 450 psi (310 N/cm^2) and a work split of 0.67, would be 2.6 percent of the total pumped flow.

In summary, the split-feed system has a reasonable flexibility in that it (1) can accommodate minor changes in component performance, system pressure drop, and topping turbine inlet temperature without redesign; (2) is more efficient than the all-bleed system in the use of hydrogen, which results in less hydrogen tankage; (3) has a potential uprating capability which is a major drawback of the all-topping system; and (4) lends itself quite readily to a parametric study of systems ranging from an all-bleed to an all-topping system. Based on these facts, a decision was made to use the split system as the reference feed system for the TWMR engine study.

REFERENCE-DESIGN SPLIT-FEED SYSTEM

This section covers the design of the reference feed system which was used with the TWMR reference engine. It includes a system description; pressure, temperature, and flow data at various stations; the physical arrangement of components; and a comparison of the reference feed system with equivalent all-bleed and all-topping systems. The reference feed system was based on the specified operating conditions given in table I, which are for a 1500-megawatt reactor. The reference design is a result of a number of iterations and cycle arrangements which followed a basic change from the initial 1725-megawatt reactor design. When the engine was derated, the division of the hydrogen pump work between the bleed and topping turbines was maintained at the previous value (i.e., one-third bleed, two-thirds topping) even though any work split was feasible at the 1500-megawatt conditions.

System Description

Figure 9(a) is a flow schematic of the reference feed system. The numerical data presented at each station are based on full-power operation. The water moderator system, also shown in this figure, is discussed in the section MODERATOR SYSTEM TURBOMACHINERY.

Tracing the flow of hydrogen leaving the storage tank, it passes through the tank shutoff valve (1V1), the gimble joint (not shown), the first-stage liquid-hydrogen pump, the second-stage pump, the propulsion nozzle where it cools the wall, and the heat exchanger where it cools the moderator. A valve (1V2) is provided to permit bypassing some of the flow around the nozzle and heat exchanger at off-design conditions. Now in a gaseous state, the hydrogen goes through a shutoff valve (2V2) to the topping turbine. This valve is required during the chilldown phase of the startup transient. A bypass valve (2V1) helps control the turbine flow. The hydrogen then cools the radiation shield and proceeds into the reactor core where most of the flow is heated to 4460° R (2480° K) and passes on through the propulsion nozzle to space. The balance, about 2.6 percent at rated conditions, is heated in four special fuel assemblies to approximately 1860° R (1030° K) and is returned to the equipment area to drive the bleed turbines. The bleed gas proceeds through three turbines in series, namely, the poison pump turbine, the water pump turbine, and the first-stage liquid-hydrogen pump turbine. Each turbine has a throttle valve (3V7, 3V1, and 3V3) and a bypass valve (3V8, 3V2, and 3V4) for pressure and flow control. The gas, now at low pressure, passes overboard through two or more auxiliary nozzles.

Figures 9(b) and (c) show the conditions throughout the reference feed system at power levels of 60 and 37 percent, respectively. The reduced power calculations were made primarily for the use of those groups analyzing engine dynamics and controls to check the accuracy of the analog and digital models that simulated the engine system.

Background information used in arriving at the values shown in the schematics of figure 9 is covered in appendixes A to C. Appendix A gives the source of pressure and temperature data. Appendix B shows how analytic representations of the turbine performance curves were developed. Appendix C discusses the sizing of valves.

Equipment Arrangement

Figure 10(a) shows the arrangement of the engine components which are external to the reactor pressure vessel. This equipment is located between the reactor vessel and the liquid-hydrogen storage tank. A radiation shadow shield located in the reactor vessel head protects the equipment from excessive radiation damage. Figure 10(b), which is section C-C of figure 10(a), shows the location of tanks required in the water and poison systems. Not shown on the drawings are the two or more auxiliary nozzles through which the bleed gas is passed overboard after it has gone through all the bleed system turbines. These nozzles can probably be used for vector control. Pneumatically operated butterfly valves were used in the design of the hydrogen and water systems because of their small size and weight. Table II lists all the valves shown in figures 9 and 10 and gives their nominal size and maximum flow coefficient. Expansion joints are included at critical points in the piping system. All equipment shown in figure 10 is directly connected to one or more of the three flow systems. The water moderator and poison systems are described in detail in the sections discussing the turbopumps for these two systems.

The hydrogen flow path can be traced in figure 10(a) by following the system description given previously. The bleed gas leaves the reactor vessel head in four 1.25-inch (3.18-cm) pipes, one for each of the four special bleed fuel elements in the core. These pipes are manifolded into a single 2-inch (5.08-cm) line which carries the flow to the poison turbine.

Figure 10(a) shows a poison-to-hydrogen heat exchanger which was added to the design in the region between the topping turbine outlet and the reactor vessel inlet. The bulk of the hydrogen flow leaving the topping turbine goes directly into the reactor vessel. The balance, about 14 percent at rated conditions, bypasses through the heat exchanger where it cools the liquid poison, and then returns to the main stream before entering the reactor vessel. The main flow is restricted somewhat to force fluid through the poison heat exchanger. The schematic drawings of figure 9 do not show this design change. Its effect on cycle calculations would be minor.

Comparison of Reference Feed System with All-Bleed and All-Topping Systems

A comparison was made of the reference-design split-feed system with an all-bleed system and with an all-topping system for the same reactor. The results are listed in table III. While the values are approximate because of several assumptions, they do point out some of the basic differences in characteristics of the systems.

The major assumptions affecting the accuracy of the all-bleed system calculation were as follows:

(1) The nozzle chamber pressure and temperature and the pressure drop and temperature rise of the hydrogen in the nozzle cooling passages were the same as in the reference feed system. Actually, a modification of the nozzle shape would be required because of the reduction in the nozzle hot gas flow rate from 90.3 to 88.0 pounds mass per second (41.0 to 40.0 kg/sec).

(2) Pressure drops in piping and other components were the same as the reference feed system.

The most significant result of the all-bleed calculation was the 5.1 percent bleed rate required compared with 2.6 percent required by the reference split system. This difference could well represent a weight saving for the split system of many tons (kg) in hydrogen alone. The decrease in overall engine specific impulse from 829 seconds for the reference feed system to 819 seconds for the all-bleed system represents a 1.2-percent loss. A minor advantage of the bleed system is the lower pump discharge pressure of 945 psia (652 N/cm^2 abs) compared with 1179 psia (802 N/cm^2 abs) for the reference feed system.

The all-topping calculation was somewhat more involved. The major assumptions were as follows:

(1) The nozzle chamber pressure and temperature and the pressure drop and temperature rise of the hydrogen in the nozzle cooling passages were the same as in the reference feed system. Actually, a modification of the nozzle shape would be required because of the increase in the nozzle hot gas flow rate from 90.3 to 92.7 pounds mass per second (41.0 to 42.1 kg/sec).

(2) The pump and turbine efficiencies were 70.8 and 70 percent, respectively, the same as the topping turbopump in the reference feed system.

(3) Hydrogen, water, and poison pumping powers were lumped together and furnished by one turbine for the topping turbine calculation with no consideration as to how these pumps would actually be arranged.

(4) Equations (B3) to (B6), developed for the reference-design topping turbine, could be used.

(5) The reference-design topping pump performance curve could be used to obtain speed as a function of pump head.

(6) The temperature entering the topping turbine was 312° R (173° K), and the temperature entering the reactor was the same as that in the reference feed system.

The all-topping-system specific impulse of 839 seconds is 1.2 percent higher than that of the reference feed system. The higher specific impulse would appear to be the most important factor in favor of the topping system, as less hydrogen would be required to accomplish a given mission. The propulsion nozzle and other components would be heavier in the topping system because the pump discharge pressure is 26 percent higher than that of the reference feed system. If uprating capability and system flexibility were not basic design aims, the all-topping system would be the logical choice for a 600-psia (414-N/cm^2 abs) chamber pressure system. It is assumed that the water and poison pump turbines could be incorporated into the topping system without encountering control problems.

FEED SYSTEM TURBOMACHINERY

The study of turbomachinery for the various feed systems proceeded in parallel with the cycle studies. Turbopumps were sized and their performance characteristics were predicted for each feed system studied. To aid in the machinery selection, certain ground rules were established after considering a number of factors which could affect the design. These considerations and the resulting ground rules are discussed first, and then the turbopumps selected for the reference feed system are presented.

General Considerations

Pumping machinery types and supports. - The general types of pumping machinery that were considered are illustrated in figure 11. The simplest bearing system, a mid-bearing support with an overhung pump and turbine, is illustrated in figure 11(a). Good bearing alinement can be maintained with this design. The turbopump can be supported either from a flange at the pump inlet or by brackets on the bearing housing. Critical clearances are easily maintained throughout the unit. This arrangement is limited to single-stage centrifugal type pumps and/or axial flow pumps and turbines with a maximum of three stages.

A straddle mounted pump with an overhung turbine is shown in figure 11(b). This arrangement allows the use of a multistage axial flow pump. The bearings, and in particular the inlet bearing, must be capable of operating in the pumped fluid. The structure for supporting the bearings, which includes a long pump housing, is complex and is exposed to a wide range of temperatures. Maintaining suitable clearances and alinement is certainly more difficult than the midbearing design. The overhung turbine arrangement is limited to three turbine stages or less.

Figure 11(c) illustrates the problems associated with straddle mounting both components, as may be necessary if a large number of turbine stages are to be utilized. In this configuration, it is necessary to balance end thrust on each component by the use of thrust balance plates, since total unbalance of each component from fluid forces may be several hundred thousand pounds force (N). The pump and turbine are connected by a torque coupling capable of only a small misalinement. In addition to the problem of supporting and maintaining alinement of each component, it is necessary to tie the two components together by a load-carrying member to maintain alinement of the pump with the turbine. The hot bearing and support at the exhaust end of the turbine is considered a major problem.

Because the arrangements shown in figures 11(a) and (b) were considered to be more feasible, the system shown in figure 11(c) was not studied further. Elimination of the straddle mounted arrangement was a controlling factor in the rest of the study since it ruled out the use of more than three turbine stages.

Pump seals. - Two general types of seals were considered, the use of which depends on the location of the hydrogen tank shutoff valve. In the first case, the shutoff valve is located between the tank and the pump suction. A good pump shaft seal is required during pump operation only, since during rocket coast periods the valve is closed. The tank valve must be relatively large since it must operate with only a small pressure drop ahead of the pump. With hydrogen as the propellant, the pump must be cooled before it is started to avoid violent pressure pulses due to vapor formations. The time and amount of propellant required for cooldown are not large, but the attainment of consistent starts and thrust buildup appeared to be a problem.

In the second case considered, the pump is stored cold (wet) with the shutoff valve located downstream from the pump. The valve can be much smaller and can operate with a larger pressure drop. The pump suction conditions are improved because of less resistance in the inlet line. The positive shaft seal required both during pump operation and shutdown periods was expected to be a major problem. Lift-off seals were under development for several turbopumps such as the M-1 axial flow liquid-hydrogen fuel pump (ref. 3), but no results were available at the time of this study. The lift-off seal is a positive contact seal when the pump is shut down and is made to lift off its seat by hydraulic force or other means to provide a small running clearance and controlled leakage when the pump is in operation. The first arrangement, with the valve located between the tank and pump, was recommended as the most suitable for the TWMR study.

Turbopump bearings. - Turbopump bearings and seals are usually the controlling factors pertaining to life expectancy or reliable operation. Thus, throughout this study, low bearing DN numbers (bearing bore diameter in millimeters times rpm $< 1.5\times10^6$) were considered desirable. Low bearing DN numbers will also assure low seal rubbing speeds. Considerable attention was also to be given to low bearing loads (radial and axial) and to proper mounting and alinement conditions.

Pump stability. - Pump flow and pressure stability are major problems in some vehicles. Of course, these are systems problems, but some of the pump parameters that may lead to a stable system have become evident. Two considerations pertaining to pump stability that were to be incorporated in this study are illustrated in figure 12. A centrifugal pump inlet designed for high suction performance (a high suction specific speed) is shown in figure 12(a). The large inlet is necessary to keep approaching fluid velocities low. As a result, there is a strong tendency for flow eddies to form at the pump inlet and within the blade passages. These eddies are not particularly stable, especially when a flow perturbation is experienced with the low approach velocity. Eddy movement is associated with, and may amplify, flow and pressure perturbations. Thus, for this study, the pumps were not to utilize the high suction performance. An upper limit of 30 000 was placed on the suction specific speed S_s even though some pumps in use have been designed for $S_s = 40\ 000$ or more.

Two pump performance characteristic curves are shown in figure 12(b). The solid curve is obtained with axial flow machines and the dashed curve is obtained with centrifugal-type pumps. With the flat (dashed) characteristic, small head perturbations result in large flow changes throughout the system, whereas the steep negative slope tends to limit system fluctuations. This is an extremely simplified comment since either could undoubtedly be made to work. It was anticipated that less difficulty would be encountered with the axial-flow-type pump.

As a result of these considerations, the study effort was to concentrate mainly on axial flow pumps and the utilization of an inducer to obtain good suction performance. An inducer is a high suction performance pump impeller, usually helical in shape, which precedes the high pressure stages of a centrifugal or axial flow pump. It is designed to operate under cavitating conditions to increase the design speed of the pump and thus keep the size and weight down.

Cavitation. - In the design of feed system turbomachinery, it is important that careful consideration be given to pump cavitation requirements. Pump suction specific speed S_s is usually set at the state-of-the-art value. However, S_s had already been set at a conservative 30 000. The equation for suction specific speed is

$$S_s = \frac{N(Q)^{1/2}}{(H_{sv} - FPE)^{3/4}} \tag{4}$$

A moderate rotational speed N was selected, and, of course, the flow rate Q was set by the specifications of table I. The value of H_{sv}, the feet of head above vapor pressure at pump suction, must now be established:

$$H_{sv} = H_g + \frac{P}{\rho} - H_L - H_v \tag{5}$$

The actual height of fluid above the pump H_g usually is the largest factor in a launch vehicle (see fig. 13). During flight, the acceleration is usually sufficient to make up for the reduction in fluid height in the tank. However, in the case being considered, where the vehicle may coast or "park" at or near zero gravity, this term may decrease to zero.

Ullage pressure P which is the pressure of the gas above the liquid, is a strong factor with low-density hydrogen. But ullage pressure is limited primarily by the weight of the tank structure, especially when a large tank volume is considered. Since a tank shutoff valve was to be considered, some head losses H_L would occur ahead of the pump even with relatively large feed lines. Fluid vapor head H_v is, of course, directly affected by propellant temperature or the heat input to the tank. Considering the positive and negative terms in the H_{sv} expression, then, for this application it would be highly desirable to assume a capability of $H_{sv} = 0$. In this event, it would be necessary that the entire denominator of the suction specific speed equation involve the fluid property effect FPE. By using the method presented in reference 4 for predicting FPE, a value of 163 feet (49.7 m) was obtained at the TWMR bleed pump inlet conditions of 40.5° R and 30 psia (22.5° K and 20.7 N/cm^2 abs).

From these considerations, the term H_{sv} = FPE was set at 163 feet (49.7 m) of fluid or 4.8 psi (3.31 N/cm^2) for this study. Pumps of this suction design could be developed to pump boiling hydrogen if necessary.

Turbines and control margins. - The machinery support considerations discussed in the section, Pumping machinery types and supports severely limited the number of full flow turbine stages that could be used. Reentry type turbines were investigated and eliminated from further consideration because of the high crossover duct losses, leakage between stages, and lack of good hot operating experience.

Although only axial flow turbines of three stages or less were to be considered, it was to be kept in mind that turbopump speeds may be severely limited when these turbines are overhung from the bearing system. The turbine rotor usually contributes heavily to the determination of critical speed since alloys that will withstand the high temperature levels are usually heavy.

An arrangement study of the turbines in the bleed system resulted in locating them in series, with a throttle valve before each turbine and a bypass line containing a bypass valve around each (see fig. 9(a) for a schematic drawing of the reference feed system). This arrangement was selected so that the bleed flow rate and pressure level in the special reactor fuel elements which heat this gas, can be held constant for a given reactor power level, regardless of relative changes in the power requirements of the turbines. All bleed system turbines were initially to be designed to operate with the inlet pressure throttled 20 percent and with 20 percent of the flow bypassed for rated full-power operation. This would allow a power margin for off-design control and acceleration and for future increases in requirements without having to change the turbine design. If the valves and/or control margins are later determined to be too large or unnecessary, the design could be optimized by reducing the valve sizes, replacing the valves with fixed orifices, or eliminating the valves altogether.

The topping turbine was to have a bypass line and valve and be initially designed to operate at rated reactor power with 20-percent flow bypassed. A valve also was to be provided at the topping turbine inlet for shutoff use only.

Component performance. - Performance margins or allowances in any system are necessary and can control the selection of turbopump components. Their use reduces the need for redesign. Thus, in all turbopump systems, conservative component efficiencies were to be utilized so that reliable performance levels could be estimated. An additional requirement was that system feasibility would still be preserved if the system pressure loss SL were increased 150 psi (103 N/cm^2). This would allow for future design changes, if required.

Design life and restart capability. - Although no specific space mission had been defined, an engine operating time was established for design purposes, and the engine was also to have a restart capability. To meet the design aims of the engine, it was decided that all turbomachinery should be capable of operating for 10 hours at rated conditions and be able to start and stop 25 times, with long periods of shutdown between runs.

Existing turbomachinery. - The development of cryogenic pumping machinery to meet the particular demands of a space propulsion system involves a large expenditure of time and effort. The possibility of minimizing this development process was to be investigated by considering the adaptability of existing design configurations of liquid-hydrogen pumping machinery to the requirements of the TWMR engine.

The use of existing hardware has several advantages and also some disadvantages. Accurate pump performance characteristics are required for studying engine system dynamics and controls. Off-design performance such as stall characteristics and H_{sv} requirements cannot be predicted accurately for theoretical designs. However, engineering judgment must be used in adapting existing equipment for TWMR use, as the design and performance data will only be reliable and realistic if the modifications required are minor. It may be necessary to compromise one or more ground rules in order to use existing hardware. In selecting turbomachinery, it should also be remembered that the materials of construction may be subjected to high levels of nuclear radiation.

Summary of ground rules. - The ground rules that were established for turbomachinery design are as follows:

(1) Only turbomachinery with a midbearing support or a straddle mounted pump with overhung turbine were to be considered. No more than three axial flow turbine stages were to be used.

(2) A tank shutoff valve was to be used between the storage tank and hydrogen pump inlet, and the pump was to have a dynamic type seal.

(3) Conservative bearing DN values ($<1.5\times10^6$) and low bearing loads were to be used.

(4) Conservative suction specific speeds ($S_s \le 30\ 000$) were to be used. The study effort was to concentrate on the use of axial flow pumps utilizing an inducer.

(5) The pump was to be designed to operate at a value of net positive suction head plus fluid property effect (H_{sv} + FPE) equal to 163 feet (49.7 m) of fluid or 4.8 psi (3.31 N/cm^2).

(6) All bleed system turbines were to be designed to operate at full power with the inlet pressure throttled 20 percent and with 20 percent of the flow bypassed. The topping turbine was to be designed to operate at full power with 20 percent of the flow bypassed.

(7) Conservative component efficiencies were to be used and the system was to be able to function if the system pressure loss SL increased 150 psi (103 N/cm^2) over the predicted value.

(8) All turbomachinery was to have a full-power operating capability of 10 hours and be able to start and stop 25 times, with long shutdown periods between runs.

(9) Existing turbomachinery was to be utilized wherever possible.

Reference Design

The reference feed system contains two hydrogen turbopumps in series, as described earlier. The first-stage pump, commonly called the bleed pump, is driven by a bleed turbine. The second-stage pump, or topping pump, is driven by a topping turbine. The rated flow rate for each pump is 92.7 pounds mass per second (42.1 kg/sec). The head requirements are 384.3 psi (265 N/cm^2) and 768.5 psi (530 N/cm^2) for the bleed and topping pumps, respectively.

The process of selecting two turbopumps to meet these requirements included a survey of existing pumping machinery. The survey indicated that, with minor changes, both the bleed and topping pumps could be based on the Mark XV fuel pump that was developed for the J-2 program (ref. 5). This pump is a flight-version liquid-hydrogen pump which has had considerable testing. A decision was made to base the design of both pumps on the Mark XV pump and to keep the same theoretical drive turbine designs that had been used earlier in the study. In making this decision, the ground rule requiring a zero net positive suction head capability was forfeited. To meet the requirements of the resulting reference design bleed pump, it was necessary to provide an H_{sv} of 102 feet (31.1 m) (NPSP = 3 psi (2.07 N/cm^2)) at the pump suction. If necessary, H_{sv} can be changed by changing the ullage pressure P (see eqs. (4) and (5)). Adding 102 feet (31.1 m) to an FPE of 163 feet (49.7 m) increased the value of the expression H_{sv} + FPE to 265 feet (80.8 m). The S_s corresponding to 265 feet (80.8 m) is 30 000 at the bleed pump rated flow and speed, as required. This compromise of an initial design aim was considered to be minor.

The final topping pump performance characteristics were based on test results of an earlier six-stage (plus inducer) version of the Mark XV pump. The final bleed pump performance characteristics were based on these same test results, but the head coefficient was scaled down to simulate a five-stage pump plus inducer. Other than a decrease in length due to decreasing the number of stages, the dimensional changes required were small. The following table gives a brief comparison of the three turbopumps:

Turbopump	Turbine			Pump				
	Number of stages	Mean blade diameter		Number of stages (a)	Blade hub diameter		Blade tip diameter	
		in.	cm		in.	cm	in.	cm
Mark XV	2	12.5	31.8	7	6.00	15.2	7.28	18.5
TWMR bleed	3	12.6	32.0	5	5.75	14.6	7.53	19.1
TWMR topping	1	10.22	26.0	6	5.915	15.0	7.367	19.5

[a]Each unit also has an inducer.

The three-stage TWMR bleed turbine is almost identical in mean blade diameter to the Mark XV two-stage turbine. Both operate on hot gas. The single-stage TWMR topping turbine is a little smaller in mean blade diameter and operates on much colder gas. Rated inlet conditions and flow rates for the TWMR turbines are as follows:

Conditions	Bleed	Topping
Inlet pressure, psia; N/cm^2 abs	146; 101	982; 677
Inlet temperature, ^{o}R; ^{o}K	1789; 994	302; 168
Flow rate, lb mass/sec; kg/sec	1.927; 0.875	73.35; 33.2

The pump, turbine, and overall unit design parameters and current operating conditions are listed in more detail in tables IV to VI. The weight (349 lb mass (158 kg)) and the moment of inertia of the rotating assembly 2.774 (in.)(lb force)(sec^2) or 0.314 (kg)(m^2) of the seven-stage Mark XV turbopump were used as the estimate for both the bleed and topping units.

The bleed pump and turbine performance curves are included in figures 14 and 15. The topping pump and turbine curves are presented in figures 16 and 17. Minor changes were periodically being made in the operating conditions for these units due to more refined calculations and design changes in components of the feed system. Thus, two points are indicated on the performance curves. One represents the initial design point on which the curves were based; the other represents the operating point at the final rated conditions. Only the final rated operating conditions are listed in tables IV and V. The initial design conditions are noted on some of the performance curves.

The turbopumps are designed to operate up to 10 percent overspeed. The H_{sv} of 102 feet (31.1 m), which was provided at the bleed pump inlet, allows for some off-design flexibility. The dashed line on the pump curves in figures 14(a) and 16(a) represents the approximate upper limit in head and flow imposed on the bleed pump by the 102 foot (31.1 m) H_{sv} (NPSP = 3 psi (2.07 N/cm^2)). The topping pump is not limited by its own H_{sv} requirement but by the bleed pump flow rate because of the series arrangement.

The density of liquid hydrogen is considered to be constant at all operating conditions for a given pump. This assumption is an approximation because the actual density varies from 4.228 pounds mass per cubic foot (67.6 kg/m^3) at the bleed pump suction to 4.515 pounds mass per cubic foot (72.3 kg/m^3) at the topping pump discharge.

The mechanical design layouts for the bleed and topping turbopumps are included as figures 18 and 19, respectively. Since the physical size and operating conditions of the turbopumps are so close to those of an existing pump, no calculations of critical speed, thrust balance, blade and disk stresses, bearing loads, etc., were performed.

The radial bearings for both hydrogen turbopumps are angular contact ball bearings. The axial thrust is taken by a self-energized balance piston. The bearings are lubricated and cooled by the hydrogen being pumped and thus require no external tubing or lubrication systems. The DN values are less than 1.5×10^6, which is well within the present state of the art. The bearings are stainless steel except for the cages which are glass-filled polytetrafluoroethylene (PTFE). The effect of nuclear radiation on PTFE was not investigated. This region of the pump may require additional shielding, or another material may have to be used for the cages.

The seals used in the hydrogen turbopumps are the positive bellows type which use a carbon graphite stationary element rubbing against a chromium plated 440-C stainless-steel ring. The seal housings are also stainless steel. Rubbing speeds are well within the state of the art.

No detailed investigation was made of pump and turbine materials. Except for possible limitations due to the nuclear environment, which is yet undefined, the materials requirements are within the present state of the art. Further investigation would be desirable, however, to determine if lighter materials such as alloys of aluminum and titanium could be used.

Several other areas should also be investigated such as the pump suction requirements during restart in space and the effect of long shutdown periods on bearings and seals when exposed to space vacuum. In connection with pump suction requirements, it would be desirable to develop a pump with a zero net positive suction head capability. Some development may be required to ensure good reliability for the 10-hour life and 25 starts and stops required.

MODERATOR SYSTEM TURBOMACHINERY

System Description

The TWMR reference engine requires a water pump to circulate the moderator through a water-to-hydrogen heat exchanger for cooling purposes. The flow schematic in figure 9(a) shows that the water flow in the reactor follows two parallel paths: 93.7 percent of the flow passes through the core, while the remaining 6.3 percent is used to cool the side reflector. All the flow then passes through the heat exchanger, the main water pump, and a check valve (4V1), and then returns to the reactor. The system contains a small motor-driven auxiliary pump in a bypass circuit around the main pump. This circuit also contains a check valve (4V2) near the auxiliary pump discharge to prevent backflow through the pump when it is not running. The auxiliary pump is used for startup and after shutdown of the reactor.

Additional details of the water system are shown in the equipment arrangement drawings (fig. 10). The water system pressurizer shown in figure 10(b) is a 20-gallon (0.0756-m^3) accumulator. It is connected to the water system inside the reactor pressure vessel. Its prime purpose is to keep the pressure in the water plenum downstream of the outlet water baffle as close to the nozzle chamber pressure as possible to minimize the pressure loading on the outlet tube sheet. The gas side of the water system pressurizer is connected to the nozzle chamber. Valves (6V1) and (6V2) in the gas line are supply and vent valves, respectively. The pressurizer also serves as an expansion or surge tank and makeup water supply, should water be lost as the result of decomposition or pump seal leakage.

Reference Design

The water pump head and flow requirements at rated conditions are 153 psi (106 N/cm^2) and 1040 pounds mass per second (472 kg/sec), respectively. Since the NPSP available is high (491 psi (338 N/cm^2)), there is no problem due to cavitation.

The rated turbine flow rate is 1.927 pounds mass per second (0.875 kg/sec), which is approximately 80 percent of the bleed flow rate. Hydrogen enters the turbine at 1860° R (1030° K) and 250 psia (172 N/cm^2 abs).

Tables IV to VI list the final rated operating conditions and design parameters in detail. A major design aim was to increase the speed as much as possible and thus reduce the size and weight of the unit, without exceeding a conservative pump suction specific speed of 6000. Figures 20 and 21 cover the water pump and turbine performance characteristics. These curves show the design point and the final operating point. The difference between the design point and final operating point is a slight increase in pump head due to water system design changes and more refined calculations.

Figure 22 is a mechanical design layout of the proposed turbopump. The two radial bearings are angular contact ball bearings that can take thrust in either direction. The DN values are conservative. The bearings are stainless steel except for bronze or monel cages and are tentatively to be lubricated and cooled by a radiation resistant fluorocarbon oil. Additional investigation of other type bearings that can be lubricated and cooled by the fluid being pumped (similar to the hydrogen pump bearings) would be most desirable. This would eliminate the auxiliary oil system required external to the pump and problems due to using oil in a nuclear environment. Also, the seal design would be simplified.

The seals used in the water pump are the same type and made of the same materials as those for the bleed and topping turbopumps discussed earlier. Rubbing speeds are conservative. The seal arrangement shown in figure 22 is based on the use of oil for bearing lubrication. Two sets of seals are used outboard of each bearing. Between each set is a vented space to prevent contamination of the oil or water. Each set consists of two seals in series which are stacked radially.

The comments regarding the materials of construction for the hydrogen turbopumps given earlier also apply to the water turbopump. No detailed study was made in this area. Reference 6 indicates that the use of carbon in the moderator system will cause corrosion of the aluminum. While carbon seals are used in the pump, it was assumed that the area in contact with the water was too small to be significant.

The volumetric flow rate through the water pump is about 80 percent as great as the flow rate through the hydrogen pumps, and thus the water pump is almost as large, physically, as the hydrogen pumps. Because the relatively low head requires only one axial stage, the water pump is not as long as the hydrogen pumps.

The intent of this study was to determine whether a turbopump for the TWMR application was feasible and to carry the design far enough to determine the approximate dimensions, weights, moment of inertia of the rotating element, performance characteristics, etc., for use in other phases of the TWMR study program. While no major problem areas are anticipated, several areas should be investigated. In addition to the areas mentioned in the hydrogen pump discussion, the generation of gas in the liquid system and its effect on pump startup and operation at zero gravity should be studied. Gas could come from fluid decomposition under nuclear radiation and from the corrosion of system materials. The effect of long shutdown periods between runs should also be studied. The study should include fluid temperature control, seal leakage, and the corrosion and welding of seal and bearing parts.

POISON SYSTEM TURBOMACHINERY

System Description

The TWMR reference engine uses a chemical poison system for reactor control. A flow diagram of this liquid system, shown in figure 23, includes all poison system components which are shown on the component arrangement drawings (fig. 10). The main poison flow enters the reactor vessel at isolation valve 5V1. It passes through an inlet plenum, proceeds in parallel through a number of control tubes which are located in the reactor core, and then through an exit plenum. The flow leaves the reactor vessel at isolation valve 5V2, passes through a heat exchanger where it is cooled by hydrogen, and then passes through the main circulating pump, a pump isolation valve 5V6, a flow-limiting orifice, an electromagnetic flowmeter (FRA-1), and back to the reactor vessel.

The poison system also contains an ion exchanger located in a bypass circuit. The ion exchanger removes poison from the circulating solution when an increase in reactor power is desired. The bypass flow leaves the main flow just downstream of valve 5V6 and passes through an overtemperature flow shutoff valve 5V7 which protects the resin. The flow then passes through the ion exchanger, a turbine-type flowmeter (FR-2), a flow control valve (5V3), and back to the main stream just upstream of valve 5V1.

A 10-gallon (0.0378-m^3) poison system pressurizer is connected to the main flow circuit just upstream of flowmeter FRA-1. This accumulator is identical in function to the water system pressurizer described previously. It maintains a pressure in the poison system close to that in the nozzle chamber to minimize the pressure differential between the poison and moderator in the core, and it acts as an expansion tank and makeup fluid supply. The gas side of the pressurizer is connected to the nozzle chamber through the same line as the water system pressurizer. Valves 6V3 and 6V4 in the gas line are supply and vent valves, respectively.

The poison system contains a small auxiliary motor-driven pump in a bypass circuit which leaves the main line near the heat exchanger outlet. This pump is used during startup and after shutdown of the reactor. The flow passes through the auxiliary pump, a pump isolation valve 5V8, and then joins the ion exchange line upstream of the ion exchanger.

Another important part of the system is the poison concentrate supply. The concentrate is stored in a 5-gallon (0.0189-m^3) reservoir similar to the pressurizer. It is pressurized with hydrogen gas regulated so that the pressure in the poison concentrate reservoir will be 50 psi (34.5 N/cm^2) above the pressure of the rest of the system. The reservoir is connected to the main stream through two parallel lines. Each line contains a flow-limiting orifice, a turbine-type flowmeter (FR-3 and FR-4), and a control valve (5V4 and 5V5). One line is for coarse control and one for fine control. The concentrate is injected into the main stream ahead of isolation valve 5V1, just before the flow reenters the reactor to provide a minimum response time for control purposes. The poison system is filled and drained through valve 5V9 which ties into the main line downstream of isolation valve 5V2. A vent valve (5V10) is located near the discharge of the main poison system circulation pump.

The chemical poison system is covered in detail in reference 7.

Reference Design

The main poison pump flow and head requirements at full power are 82.5 pounds mass per second (37.5 kg/sec) and 233 psi (161 N/cm^2), respectively. For pumping purposes, the poison solution can be considered the same as water. The density is 61.7 pounds mass per cubic foot (988 kg/m^3) at the rated pump inlet conditions of 585° R (325° K) and 367 psia (253 N/cm^2 abs). As was the case with the water pump, no cavitation problem exists at rated conditions since the NPSP available is sufficiently high (365 psi (252 N/cm^2)).

The poison pump drive turbine is located first in the series of bleed system turbines (see fig. 9(a)). The hydrogen conditions at the turbine inlet for full reactor power are 1871° R (1040° K) and 488 psia (336 N/cm^2 abs), and the rated turbine flow rate is 2.16 pounds mass per second (0.98 kg/sec). Ten percent of the bleed flow is bypassed around the turbine. The change from the original 20-percent bypass was made necessary by an increase in pump head requirement.

Figure 24 is a mechanical design layout of the reference-design poison turbopump. Operating conditions and design details for this unit are presented in tables IV to VI. The poison pump performance characteristics are given in figure 25, and the turbine characteristics are given in figure 26.

The poison flow required at reduced reactor power levels has not, as yet, been well defined because of the significant interaction between the poison system requirements and the final results of the overall rocket system control study (ref. 8). Since the response time of the poison system is directly proportional to the poison circulation rate, it was decided for the purposes of the study herein reported, to take the conservative approach and maintain the flow rate as close to the full-power flow rate as possible. Calculations indicated that it was possible to maintain the rated flow rate of 82.5 pounds mass per second (37.5 kg/sec) for the 100- and 60-percent reactor-power cases. However, in the 37-percent reactor-power case, the poison flow rate was limited to 69.9 pounds mass per second (31.4 kg/sec) because of a lack of turbine power. For this case, the pump suction pressure was 52 psia (35.8 N/cm^2 abs), based on a pump head requirement of 167 psi (115 N/cm^2) and a chamber pressure of 219 psia (151 N/cm^2 abs). This pressure is just within the net positive suction pressure limit of 50 psi (34.5 N/cm^2) required by the pump for this flow.

It is apparent that in addition to a turbine power limitation at reactor powers less than 37 percent, there would also be an NPSP limitation which would prevent obtaining a poison flow rate of 82. 5 pounds mass per second (37. 5 kg/sec). Since the NPSP requirement varies approximately with the square of the flow rate, it would be desirable to control the poison pump suction pressure so that it never drops below 50 psia (34. 5 N/cm^2 abs) at low reactor power levels. This would ensure that the solution does not boil and that there is no in-leakage of turbine gas or lubricant past the shaft seals. The poison system is designed to withstand an internal pressure up to 600 psia (414 N/cm^2 abs) so that any minimum poison system pressure level could be used below this value.

The poison turbopump uses the same type of bearings and bearing materials as the moderator turbopump. The shaft seals are the same type and made of the same materials as those of the hydrogen turbopumps.

The requirements of the auxiliary pump, located in parallel with the main poison pump, are not presently well defined. As a result, the auxiliary pump design has not been investigated. It is expected that the flow rate will be about 10 percent that of the main pump.

Many of the design requirements of the poison turbopump are the same as, or similar to, the water turbopump. No major problem areas were apparent during the study. Areas that will require further investigation are the same as those for the water pump.

CONCLUSIONS

The study of feed systems and turbomachinery for the TWMR reference engine led to the following conclusions:

1. Topping systems are limited by nozzle chamber pressure. This limit occurs in the vicinity of 600 to 800 psia (414 to 552 N/cm^2 abs) for TWMR type engines where the available turbine inlet temperature is near 300° R (167° K).

2. Bleed rates required by bleed systems are high and increase with chamber pressure. Typical rates for TWMR type engines are 5.1 and 5.7 percent for chamber pressures of 600 and 800 psia (414 and 552 N/cm^2 abs), respectively.

3. Basic feed system types can be combined to overcome limitations and undesirable features.

4. A split-feed system is less sensitive to minor changes in component efficiency, turbine inlet temperature, and system pressure loss than a bleed or topping system. The split system is also more flexible in how it can accommodate such changes.

The design of turbomachinery to meet the hydrogen pumping requirements of the reference split-feed system should not be a problem since similar hardware has already been developed for the J-2 engine. Areas that require further investigation or development are a zero net positive suction head capability; improved reliability for the 10-hour-life and 25-start-stop requirements; the effect of radiation on construction materials; and the effect of space vacuum on bearings and seals during long shutdown periods.

No major problem areas were uncovered in the study of the water and poison turbopumps for the reference engine. However, a better bearing system should be developed that could be lubricated by the fluid being pumped. Other areas requiring further investigation are gas generation from radiation and corrosion and its effect on pump startup and operation at zero gravity; the effect of radiation on component materials; fluid freezing, seal leakage, and the corrosion and welding of seals and bearings during long shutdown periods.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, July 19, 1967,
122-28-02-04-22.

APPENDIX A

FEED SYSTEM COMPONENT PRESSURE AND TEMPERATURE BACKGROUND

Table VII is a summary of the calculated pressure drops for the various components which make up the topping portion of the reference feed system. It should be pointed out that the feed system pressure drops continually changed as design changes and refined calculations were made in various components. More accurate data now indicate that the full-power system pressure drop of 1188 psi (819 N/cm^2) will increase about 4 percent (50 psi (34.5 N/cm^2)). A large part of this increase, 21.5 psi (14.8 N/cm^2), is in the reactor core drop and is discussed subsequently. The addition of a liquid poison-to-hydrogen heat exchanger downstream of the topping turbine will add about 14 psi (9.6 N/cm^2), and an increase in the drop through the reactor top reflector will add another 5 psi (3.45 N/cm^2). These losses are expected to increase the topping turbine pressure drop about 9.5 psi (6.55 N/cm^2) after work matching. These changes are minor and in no way affect the conclusions drawn from the study.

Throughout the feed system calculations, the fluid was assumed to be para-hydrogen, and the fluid properties were obtained from a digital computer code described in reference 9. Other assumptions on which table VII and figure 9 were based are as follows: The 4460° R (2480° K) chamber pressure specified in table I for full power was also used for the 60- and 37-percent reactor power levels studied. The hydrogen mass flow rate was assumed to vary linearly with reactor power level. The main liquid hydrogen storage tank pressure was assumed to be constant at 35 psia (24.2 N/cm^2 abs) for all power levels. Unless stated otherwise, the values of pressure drop for reduced power conditions were scaled down from the calculated values for 100-percent power according to the following equation:

$$\Delta p_2 = \Delta p_1 \left(\frac{\dot{w}_2}{\dot{w}_1}\right)^a \left(\frac{\rho_1}{\rho_2}\right)^b \left(\frac{p_1}{p_2}\right)^c \left(\frac{T_2}{T_1}\right)^d \tag{A1}$$

The exponents used in this equation are given in table VIII. The density ratio was only used in those parts of the system containing liquid hydrogen, and the pressure and/or temperature ratios were used for gaseous hydrogen.

The sources of component pressure and temperature used in the feed system calculations are given as follows:

Propulsion nozzle - thrust side. - The chamber pressure for 100 percent power was specified in table I.

Propulsion nozzle - coolant side. - The pressure drop and temperature rise data for all power levels came from a nozzle design study covered in detail in reference 6. The data were based on a nozzle area ratio of 40.

Heat exchanger. - The pressure drop and temperature rise data for all power levels came from a heat exchanger design study, also covered in detail in reference 6. The data were based on a heat exchanger tube length of 2.750 feet (83.9 cm).

Reactor core. - The friction and momentum pressure drops in the core fuel stages were calculated for all three power levels. The calculation of interstage losses for 100-percent power assumed that a 50-percent expansion and contraction took place. A later calculation was based on a 100-percent expansion and contraction, which accounts for most of the 21.5 psi (14.8 N/cm^2) increase in the core pressure drop discussed previously. The actual percent expansion and contraction most likely lies between the two assumptions.

Reactor top reflector. - The friction and momentum pressure drops were estimated for the holes in an unfueled beryllium plug at the top of the center fuel assembly, which is the worst case. The Mach number assumed within the holes for 100-percent power was 0.165. The entrance and exit losses were calculated at 100-percent power based on rounded edges at the inlet and outlet of each hole.

Reactor radiation shield. - Friction and momentum pressure drops through the holes in the radiation shadow shield at the top of the reactor pressure vessel were estimated for 100-percent power assuming a temperature rise of 6 R$^{\circ}$ (3.3 K$^{\circ}$) and a Mach number of 0.2. The shield entrance and exit losses were also based on well rounded holes at the inlet and outlet to keep the losses to a minimum. The pressure drop around the poison tubes and poison header was assumed negligible.

Piping losses. - A pressure drop of 5 psi (3.45 N/cm^2) was allowed between the storage tank and the first-stage liquid-hydrogen pump inlet at full power. The pump required 3 psi (2.07 N/cm^2) net positive suction pressure which means that the saturation pressure corresponding to the pump inlet temperature would have to be 27 psia (18.6 N/cm^2 abs) giving 40.5° R (22.5° K) for the tank and pump inlet temperature. The 5-psi (3.45-N/cm^2) pressure drop between tank and pump was allowed to permit a change in design to another type of tank shutoff valve without redesigning the bleed pump. Actually, this pressure drop would only be 0.5 psi (0.345 N/cm^2) if it were based on the 8-inch (20.3-cm) butterfly valve that was used in the reference engine. Thus, the tank pressure could be reduced to 30.5 psia (21.0 N/cm^2 abs). No credit has been included in calculating the NPSP available for pressure due to acceleration of the rocket under power operation. Also, no study has been made of how the storage tank would be maintained at a given pressure above saturation pressure.

Full-power pressure drops for the rest of the feed system piping (both topping and bleed portions) were calculated by conventional methods such as those given in reference 10. The feed system was broken into a number of sections so that flow, pressure, and temperature conditions were approximately the same within any one section. It was assumed that the liquid or gas was incompressible in each section. The error introduced was insignificant except possibly at the low pressure levels at the end of the bleed system. The Mach number of the gas in each section did not exceed 0.3 except at the entrance or exit of some valves and in several lines at the low pressure end of the bleed system where density and pressure drop in psi (N/cm^2) were low, and line sizes were large.

The full-power pressure drops for each section were scaled down for reduced powers. The exponent a in equation (A1) varied from 1.8 to 2.0 depending on how much of the pressure drop was pipe friction and how much was square loss in reducers, etc.

Turbines. - The values for temperature and pressure drop across all four turbines were obtained from the respective pump and turbine work-matching calculations for all three power conditions. Characteristic data required for these calculations were obtained from the pump and turbine performance curves. Analytic representations of the turbine performance curves were developed to simplify the calculations, and these equations are discussed in appendix B.

Pumps. - The total pressure rise across both hydrogen pumps plus the 35-psia (24.2-N/cm^2 abs) tank pressure must equal the total topping system pressure drop, as shown in table VII. The pump work was split so that the bleed pump furnished one-third of the total pump head required, and the topping pump the balance. The pump head requirements were determined from an iterative topping turbine and pump work-matching procedure for all three power levels. The temperature rise across each hydrogen pump was arbitrarily assumed to be about 60 percent of the theoretical rise to be more in agreement with test experience.

Valves. - Valve 1V2, which bypasses the propulsion nozzle cooling passages and the heat exchanger (see fig. 9), was assumed to be closed, and the tank shutoff valve (1V1) and the topping turbine shutoff valve (2V2) were assumed to be wide open for all three power levels. The pressure drops across the other hydrogen control valves deviate from the initial values established in the ground rules because of many changes in the components and the system operating requirements. The control valves are discussed in more detail in appendix C.

APPENDIX B

ANALYTIC REPRESENTATION OF TURBINE PERFORMANCE CURVES

Analytic representations of the turbine performance curves (figs. 15, 17, 21, and 26) were developed to facilitate work matching and the investigation of off-design operating characteristics of the TWMR.

Based on nozzle theory, the flow parameter for a given turbine can be represented as follows for a single stage (ref. 11, p. 172):

$$\frac{\dot{w}\sqrt{T_1}}{P_1} = C_x \left\{ \left(\frac{P_2}{P_1}\right)_s^{2/\gamma} \left[1.0 - \left(\frac{P_2}{P_1}\right)_s^{(\gamma-1)/\gamma}\right] \right\}^{1/2} \tag{B1}$$

Where there is more than one stage and the stage pressure ratios are assumed to be equal, the overall pressure ratio can be used as follows to give the stage ratio

$$\left(\frac{P_2}{P_1}\right)_s = \left(\frac{P_2}{P_1}\right)_o^{1/n} \tag{B2}$$

where n is the effective number of stages.

Substituting equation (B2) into (B1) gives the expression for the flow parameter for n stages in the following form:

$$\frac{\dot{w}\sqrt{T_1}}{P_1} = C_x \left[\left(\frac{P_2}{P_1}\right)_o^{1/n\gamma} \right] \left[1.0 - \left(\frac{P_2}{P_1}\right)_o^{(\gamma-1)/n\gamma}\right]^{1/2} \tag{B3}$$

This equation holds until the onset of choked flow at any stage.

The torque parameter $\tau/\dot{w}\sqrt{T_1}$ can be expressed as follows:

$$\frac{\tau}{\dot{w}\sqrt{T_1}} = C_y \left[1.0 - \left(\frac{P_2}{P_1}\right)_o^{(\gamma-1)/\gamma}\right]^{1/2} - C_z \frac{N}{\sqrt{T_1}} \tag{B4}$$

This relation, which indirectly includes an estimate of turbine efficiency, is developed in reference 12 and is based on the isentropic velocity in an ideal nozzle (ref. 11, p. 168).

The values of the constants in equations (B1), (B3), and (B4) for the TWMR turbines were derived by fitting the equations to the turbine performance curves and assuming γ to be 1.376. These values and the effective number of stages are given in the following table:

Turbine	Effective number of stages, n	Constants in eqs. (B1), (B3), and (B4)		
		C_x	C_y	C_z
Bleed	3	1.54×10^{-2}	22.5	8.3×10^{-3}
Topping	2	5.35×10^{-2}	8.96	9.58×10^{-4}
Water pump	2	1.14×10^{-2}	16.71	4.23×10^{-3}
Liquid poison pump	1	1.16×10^{-2}	4.76	2.83×10^{-4}

The topping turbine torque parameter actually requires two equations depending on the values of P_1/P_2 and $N/\sqrt{T_1}$ being considered. Equation (B4) is valid except where both $P_1/P_2 < 1.5$ and $N/\sqrt{T_1} > 800$.

In the range where both $P_1/P_2 < 1.5$ and $N/\sqrt{T_1} > 800$, equation (B4) was modified to give a better curve fit, resulting in the following equations for the topping turbine torque parameter:

$$\frac{\tau}{\dot{w}\sqrt{T_1}} = \left(\frac{\tau}{\dot{w}\sqrt{T_1}}\right)_* - \frac{\left(\frac{\tau}{\dot{w}\sqrt{T_1}}\right)_* + 2.7864}{5140}\left(\frac{N}{\sqrt{T_1}} - 800\right) \qquad \text{(B5)}$$

where

$$\left(\frac{\tau}{\dot{w}\sqrt{T_1}}\right)_* = 8.96\left[1 - \left(\frac{P_2}{P_1}\right)_o^{(\gamma-1)/\gamma}\right]^{1/2} - 9.58\times10^{-4}(800)$$

$$= 8.96\left[1 - \left(\frac{P_2}{P_1}\right)_o^{(\gamma-1)/\gamma}\right]^{1/2} - 0.7664$$

and where

$$\left(\frac{\tau}{\dot{w}\sqrt{T_1}}\right)_*$$

is evaluated at

$$\frac{N}{\sqrt{T_1}} = 800$$

The temperature ratio across a turbine can be represented closely by the following equation:

$$\frac{T_2}{T_1} = 1.0 - \frac{1.347\times10^{-4}}{c_p}\left(\frac{\tau}{\dot{w}\sqrt{T_1}}\right)\left(\frac{N}{\sqrt{T_1}}\right) \tag{B6}$$

Equation (B6) was derived from the following power balance:

$$\dot{w}c_p(T_1 - T_2) = \frac{2\pi N\tau}{60(778)}$$

APPENDIX C

CONTROL VALVES

The flow coefficient C_V used to relate flow and pressure drop in the control valves used in the TWMR reference engine, is defined as the quantity of water in U.S. gallons per minute (multiply by 3.78×10^{-3} to obtain m^3/min) at 14.7 psia (10.1 N/cm^2 abs) and 520° R (289° K), that will flow through a given valve with a pressure drop of 1 psi (0.689 N/cm^2). The rated (maximum) flow coefficient for a given valve is obtained when the valve is wide open and is denoted by $C_{V,max}$. Although determined with water, C_V may be used to relate gas flow rate and pressure drop with the use of an appropriate gas equation. Many formulas are being used to calculate C_V as discussed in reference 13. The specific formulas used in this study are conservative in that they tend to oversize the valves by basing gas density on the exit conditions. The formulas used for gaseous hydrogen are as follows:

Unchoked flow, $p_2/p_1 > 0.53$:

$$C_V = 125.3\,\dot{w}\sqrt{\frac{T_1}{p_2(p_1 - p_2)}} \tag{C1}$$

Choked flow, $p_2/p_1 \leq 0.53$:

$$C_V = \frac{250.6\,\dot{w}\sqrt{T_1}}{p_1} \tag{C2}$$

The value of C_V varies with valve size, type, and design and can only be determined by test. Nominal values of $C_{V,max}$ used in this study for sizing valves are typical for cryogenic butterfly valves. They are as follows:

Valve size		Nominal value of maximum flow coefficient, $C_{V,max}$
in.	cm	
1/2	1.27	12.5
1	2.54	50
2	5.08	200
4	10.2	1000
6	15.2	2000
8	20.3	3500

The C_V values obtained from equations (C1) and (C2), based on the pressures and temperatures in figure 9, are shown in the first three columns of table IX. The maximum C_V value expected for each of the valves is given in the fourth column of table IX and is based on the maximum value in the first three columns with an additional 25-percent margin to allow for uncertainty in the valve requirements at off-design conditions. These maximum C_V values and the preceding table were used to size the valves. The $C_{V,max}$ of the valve should be equal to or larger than the $C_{V,max}$ expected.

The topping turbine shutoff valve (2V2) and the heat exchanger and nozzle bypass valve (1V2) were added to the design because the system dynamics studies discussed in reference 8 indicated that their inclusion would greatly facilitate engine control during transient operations.

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TABLE I. - SPECIFIED OPERATING CONDITIONS

Operating conditions	Design	
	Initial	Reference
Reactor power, MW	1725	1500
Nozzle chamber pressure, psia; N/cm^2 abs	800; 552	600; 414
Nozzle chamber temperature, ^oR; ^oK	4460; 2480	4460; 2480
Hydrogen flow rate, lb mass/sec; kg/sec	107; 48.6	92.7; 42.1
Liquid hydrogen storage tank pressure, psia; N/cm^2 abs	35; 24.2	35; 24.2

TABLE II. - VALVE DATA FOR FLOW SYSTEMS[a]

Valve designation	Application	Nominal size		Flow coefficient, $C_{V,max}$
		in.	cm	
1V1	Main hydrogen tank shutoff	8	20.3	3500
1V2	Heat exchanger and nozzle bypass	6	15.2	2000
2V1	Topping turbine bypass	4	10.2	1000
2V2	Topping turbine shutoff	6	15.2	2000
3V1	Water pump turbine throttle	$1\frac{1}{2}$	3.81	115
3V2	Water pump turbine bypass	$1\frac{1}{2}$	3.81	115
3V3	Bleed turbine throttle	$2\frac{1}{2}$	6.35	300
3V4	Bleed turbine bypass	2	5.08	200
3V7	Poison pump turbine throttle	$1\frac{1}{4}$	3.18	80
3V8	Poison pump turbine bypass	$1\frac{1}{4}$	3.18	80
4V1	Main water pump check valve	6	15.2	----
4V2	Auxiliary water pump check valve	2	5.08	----
5V1	Reactor poison inlet block valve	$2\frac{1}{2}$	6.35	----
5V2	Reactor poison outlet block valve	$2\frac{1}{2}$	6.35	----
5V3	Ion exchanger poison flow control	$\frac{1}{2}$	1.27	----
5V4	Poison concentrate injection-fast control	$\frac{1}{4}$	.635	----
5V5	Poison concentrate injection-slow control	$\frac{1}{4}$	.635	----
5V6	Main poison pump isolation	$2\frac{1}{2}$	6.35	----
5V7	Ion exchanger overtemperature poison flow shutoff	$\frac{1}{2}$	1.27	----
5V8	Auxiliary poison pump isolation	$\frac{1}{2}$	1.27	----
5V9	Poison system fill or drain	$\frac{1}{2}$	1.27	----
5V10	Poison system vent	$\frac{1}{2}$	1.27	----
6V1	Water pressure regulation gas supply valve	$\frac{3}{8}$	.95	----
6V2	Water pressure regulation gas vent valve	$\frac{3}{8}$	.95	----
6V3	Poison pressure regulation gas supply valve	$\frac{3}{8}$	.95	----
6V4	Poison pressure regulation gas vent valve	$\frac{3}{8}$	.95	----

[a]See figs. 9 and 10.

TABLE III. - COMPARISON OF SPLIT-FEED SYSTEM WITH ALL-BLEED AND ALL-TOPPING SYSTEMS

Item	Feed system		
	Reference design split	All bleed	All topping
Chamber pressure, psia; N/cm^2 abs	600; 414	600; 414	600; 414
Total hydrogen flow, lb mass/sec; kg/sec	92.7; 42.1	92.7; 42.1	92.7; 42.1
Bleed flow, lb mass/sec; kg/sec	2.4; 1.09	4.73; 2.15	0
Flow through propulsion nozzle, lb mass/sec; kg/sec	90.3; 41.0	87.97; 39.9	92.7; 42.1
Bleed rate, percent	2.6	5.1	0
Hydrogen pump horsepower, hp; MW	9293; 6.93	7180; 5.35	11 680; 8.71
Water pump horsepower, hp; kW	884; 659	884; 659	884; 659
Poison pump horsepower, hp; kW	137; 102	137; 102	137; 102
Total pump horsepower, hp; MW	10 314; 7.69	8201; 6.11	12 701; 9.47
$\frac{\text{Hydrogen pump horsepower}}{\text{Reference design hydrogen pump horsepower}} \times 100$, percent	100	77	125.6
Maximum system pressure (at hydrogen pump discharge), psia; N/cm^2 abs	1179; 812	945; 652	1487; 1025
Overall engine specific impulse (vacuum), sec	829	819	839
Uprating capability	Good	Good	Poor

TABLE IV. - PUMP RATED OPERATING CONDITIONS AND DESIGN PARAMETERS

Conditions and design parameters	Pump			
	Bleed	Topping	Water	Poison
Rated operating conditions:				
Fluid	Liquid hydrogen	Liquid hydrogen	Water	Water solution
Flow:				
lb mass/sec; kg/sec	92.7; 42.1	92.7; 42.1	1040; 472	82.5; 37.5
gal/min; m^3/min	9789; 37.0	9675; 36.6	7733; 29.2	600; 2.265
Pressure rise:				
psi; N/cm^2	384.3; 265	768.5; 530	153; 106	233; 161
ft; m	13 010; 3970	25 760; 7850	365; 111	544; 166
Speed, rpm	19 910	24 900	13 254	16 350
Efficiency, percent	69.3	70.8	78.1	59.6
Power input, hp; kW	3166; 2360	6127; 4570	884; 659	137; 102
Inlet pressure, psia; N/cm^2 abs	30; 20.7	411; 283	502; 346	367; 253
Inlet temperature, ^{o}R; ^{o}K	40.5; 22.5	44; 24.5	654; 363	585; 325
Fluid density, lb mass/cu ft; kg/m^3	4.25; 68.0	4.30; 68.9	60.4; 966	61.7; 988
NPSP available, psi; N/cm^2	3; 2.07	368; 254	491; 338	365; 252
Design parameters:				
Type	Axial	Axial	Axial	Centrifugal
Number of stages	5 plus inducer	6 plus inducer	1	1
Pump (housing) inlet diameter, in.; cm	7.56; 19.2	7.397; 18.8	6.060; 15.4	2.71; 6.88
Blade hub diameter, in.; cm	5.75; 14.6	5.915; 15.0	3.9; 9.9	1.356; 3.44
Blade outlet diameter, in.; cm	--------------	--------------	-----------	4.64; 11.8
Pump weight, lb mass; kg	--------------	--------------	88; 40	62; 27.7
Fluid volume, cu in.; cm^3	--------------	--------------	660; 10 800	69; 1130
Wetted surface area, sq in.; cm^2	--------------	--------------	970; 6260	298; 1920
Bearings:				
Type	Angular contact ball bearings; self-energized balance piston for thrust		Angular contact ball bearings take thrust both directions	Matched pair angular contact ball bearings on pump end for thrust; roller bearings on turbine end
Lubrication and cooling	Liquid hydrogen from pump		Fluorocarbon oil system	
Materials	Races and balls 440-C stainless steel; cages glass-filled Teflon		Races and balls 440-C stainless steel; cages bronze or monel	
DN value	Less than 1.5×10^6			
Seals:				
Type	Positive welded bellows			
Materials	Stationary element carbon graphite against rotating chromium plated 440-C stainless-steel ring; bellows 347 stainless steel; housing stainless steel			
Arrangement	See ref. 5		Two sets of seals with vent between to prevent contamination on both pump and turbine ends; each set contains two seals in series	

TABLE V. - TURBINE RATED OPERATING CONDITIONS AND DESIGN PARAMETERS

Conditions and design parameters	Turbine			
	Bleed	Topping	Water	Poison
Rated operating conditions:				
Fluid	Hydrogen gas	Hydrogen gas	Hydrogen gas	Hydrogen gas
Flow through turbine, lb mass/sec; kg/sec	1.927; 0.875	73.35; 33.2	1.927; 0.875	2.16; 0.98
Pressure ratio	6.309	1.2809	1.529	1.0547
Speed, rpm	19 910	24 900	13 254	16 350
Efficiency (approximately), percent	45	70	45	45
Power output, hp; kW	3166; 2360	6127; 4570	884; 659	137; 102
Inlet pressure, psia; N/cm^2 abs	146; 101	982; 677	249; 171	488; 336
Inlet temperature, ^oR; ^oK	1789; 994	302; 168	1860; 1030	1871; 1040
Outlet temperature, ^oR; ^oK	1472; 818	286; 159	1771; 984	1859; 1029
Mean blade speed, ft/sec; m/sec	1095; 334	1110; 338	694; 212	432; 132
Torque, ft-lb force; N-m	835; 1190	1292; 1760	350; 476	43.9; 59.7
Design parameters:				
Type	Impulse	Reaction	Impulse	Impulse
Number of stages	3	1	2	1
Blade mean section diameter, in.; cm	12.6; 32.0	10.22; 26.0	12; 30.5	6.05; 15.4
Blade height, in.; cm:				
First stage	0.288; 0.731	1.03; 2.62	0.38; 0.965	0.577; 1.47
Second stage	------------	------------	0.43; 1.09	------------
Third stage	1.325; 3.36	------------	------------	------------
Blade axial chord, in.; cm	1.0; 2.54	0.75; 1.90	0.38; 0.965	------------
Rim width, in.; cm	1.5; 3.81	1.1; 2.795	0.9; 2.29	------------
Wheel web thickness, in.; cm	1.2; 3.05	0.8; 2.03	1.4; 3.6	------------
Wheel maximum thickness, in.; cm	2.4; 6.1	1.6; 4.06	2.8; 7.11	------------
Maximum torque, ft-lb force; N-m	2211; 3000	3422; 4660	792; 1080	------------
Shaft diameter (15 000-psi (10 340-N/cm^2) shear), in.; cm	2.08; 5.28	2.41; 6.12	1.48; 3.76	------------
Turbine weight, lb mass; kg	------------	------------	128; 581	70; 31.8
Fluid volume, cu in.; cm^3	------------	------------	320; 5250	95; 1560

[redacted]

TABLE VI. - TURBOPUMP UNIT DESIGN PARAMETERS

Design parameters	Turbopump			
	Bleed	Topping	Water	Poison
Weight, total unit (dry), lb mass; kg	[a]349; 158	[a]349; 158	216; 98	132; 59.9
Weight, rotating parts, lb mass; kg	-------------	-------------	60; 27.2	21; 9.53
Weight, stationary parts, lb mass; kg	-------------	-------------	156; 70.8	111; 50.4
Moment of inertia of rotating parts, (in.)(lb force)(sec^2); (kg)(m^2)	[a]2.774; 0.314	[a]2.774; 0.314	2.0; 0.226	0.085; 0.00962
Overall unit length, in.; cm	26.5; 67.3	23.2; 59.0	17.75; 45.1	18; 45.7
Maximum diameter (turbine), in.; cm	17.7; 45.0	15; 38.1	15.75; 40.0	9; 22.9

[a]Approximate.

[redacted]

TABLE VII. - REFERENCE FEED SYSTEM PRESSURE DROPS (TOPPING PORTION)

Component	Rated power, percent					
	100		60		37	
	Pressure drop, Δp					
	psi	N/cm^2	psi	N/cm^2	psi	N/cm^2
Propulsion nozzle:						
Thrust side	600.0	413.7	360.0	248.2	219.4	151.3
Coolant side:						
Tube friction and momentum	131.8	90.8	87.5	60.3	63.7	43.9
Entrance	8.7	6.0	3.4	2.3	1.9	1.3
Exit	5.0	3.5	3.4	2.3	3.3	2.3
Heat exchanger:						
Tube friction and momentum	16.5	11.4	11.8	8.1	8.3	5.7
Entrance	2.4	1.7	1.5	1.0	.9	.6
Exit	10.0	6.9	6.5	4.5	3.7	2.6
Reactor core:						
Fuel stages friction and momentum	85.0	58.6	56.1	38.6	38.5	26.6
Expansion and contraction	19.5	13.4	11.7	8.1	7.1	4.9
Reactor top reflector:						
Hole friction and momentum	8.0	5.5	5.3	3.7	3.5	2.4
Entrance	.7	.5	.4	.3	.3	.2
Exit	4.3	3.0	2.6	1.8	1.5	1.0
Reactor radiation shield:						
Hole friction and momentum	23.3	16.1	15.3	10.6	10.1	7.0
Entrance	.9	.6	.5	.3	.3	.2
Exit	6.3	4.3	3.8	2.6	2.3	1.6
Reactor vessel 8-in. (20.3-cm) nozzles:						
Hydrogen inlet	11.5	7.9	6.8	4.7	4.1	2.8
Hydrogen outlet	6.0	4.1	3.9	2.7	2.3	1.6
Piping losses:						
Tank to first-stage pump inlet	5.0	3.5	2.0	1.4	.8	.6
First-stage pump to second-stage pump	3.6	2.5	1.3	.9	.6	.4
Second-stage pump to propulsion nozzle	8.9	6.1	3.6	2.5	1.5	1.0
Heat exchanger to topping turbine	8.3	5.7	4.6	3.2	2.6	1.8
Topping turbine to reactor	7.0	4.8	3.4	2.3	1.9	1.3
Topping turbine	215.1	148.3	89.4	61.6	40.2	27.7
Total topping system pressure drop	1187.8	818.9	684.8	472.0	418.8	288.8
Tank pressure	35.0	24.2	35.0	24.2	35.0	24.2
Total pump head (first- and second-stage pumps)	1152.8	794.7	649.8	447.8	383.8	264.6

[redacted]

TABLE VIII. - VALUES FOR EXPONENTS USED IN PRESSURE DROP SCALING EQUATION (A1)

Component	Exponent			
	a	b	c	d
Propulsion nozzle:				
Thrust side	1.00	0	0	0
Coolant side:				
Entrance	1.85	1	0	0
Exit	1.85	0	1	1
Heat exchanger:				
Entrance and exit	2.00	0	1	1
Reactor core:				
Expansion and contraction	1.00	0	0	0
Reactor top reflector:				
Hole friction and momentum	1.80	0	1	0
Entrance and exit	2.00	0	1	0
Reactor radiation shield:				
Hole friction and momentum	1.80	0	1	0
Entrance and exit	2.00	0	1	0
Reactor vessel 8-in. (20.3-cm) nozzles:				
Hydrogen inlet	2.00	0	1	0
Hydrogen outlet	2.00	0	1	1
Piping losses:				
All-liquid lines	(a)	1	0	0
All-gas lines	(a)	0	1	1

[a]Exponent "a" varied for each section of line from 1.8 to 2.0 depending on how much pressure drop was pipe friction and how much was square loss in reducers, etc.

TABLE IX. - CONTROL VALVE FLOW COEFFICIENTS

Valve	Power level, percent			Expected maximum flow coefficient, $C_{V,max}$ (a)
	100	60	37	
	Calculated flow coefficient, C_V			
Topping turbine bypass	102	229	379	474
Water pump turbine throttle	47.3	56.2	62.5	78.2
Water pump turbine bypass	[b]11.7	37.1	49.6	62.0
Bleed turbine throttle	241	93.3	98.8	301
Bleed turbine bypass	[b]32.9	[b]61.5	[b]73.7	92.1
Poison turbine throttle	65.1	53.6	50.0	81.4
Poison turbine bypass	5.34	0	0	6.67

[a]The $C_{V,max}$ expected is the value used to size the valve. It is equal to the largest value in the first three columns plus an additional 25-percent margin.

[b]These valves are choked.

[redacted]

[redacted]

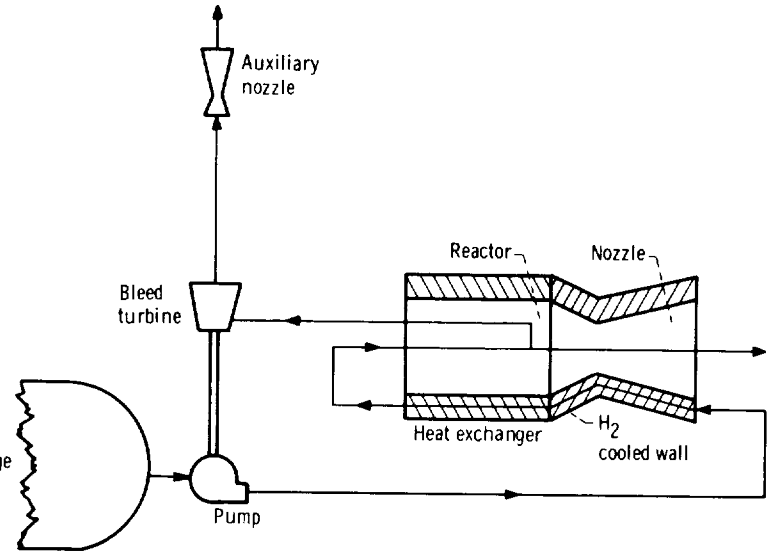
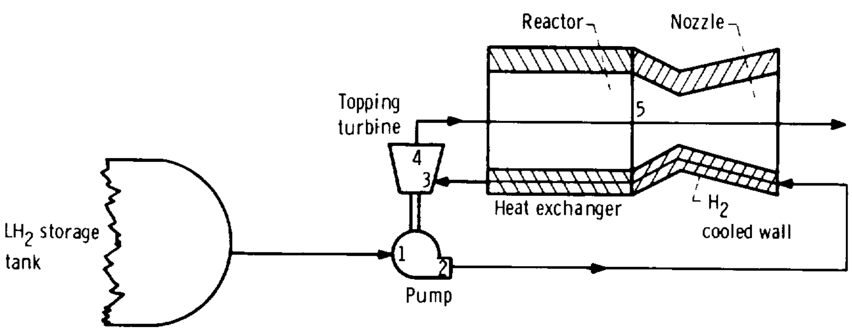


Figure 1. - Schematic drawing of hot bleed system.



Flow station	Location
1	Pump inlet
2	Pump discharge
3	Topping turbine inlet
4	Topping turbine outlet
5	Propulsion nozzle chamber

CS-38996

Figure 2. - Schematic drawing of topping system.

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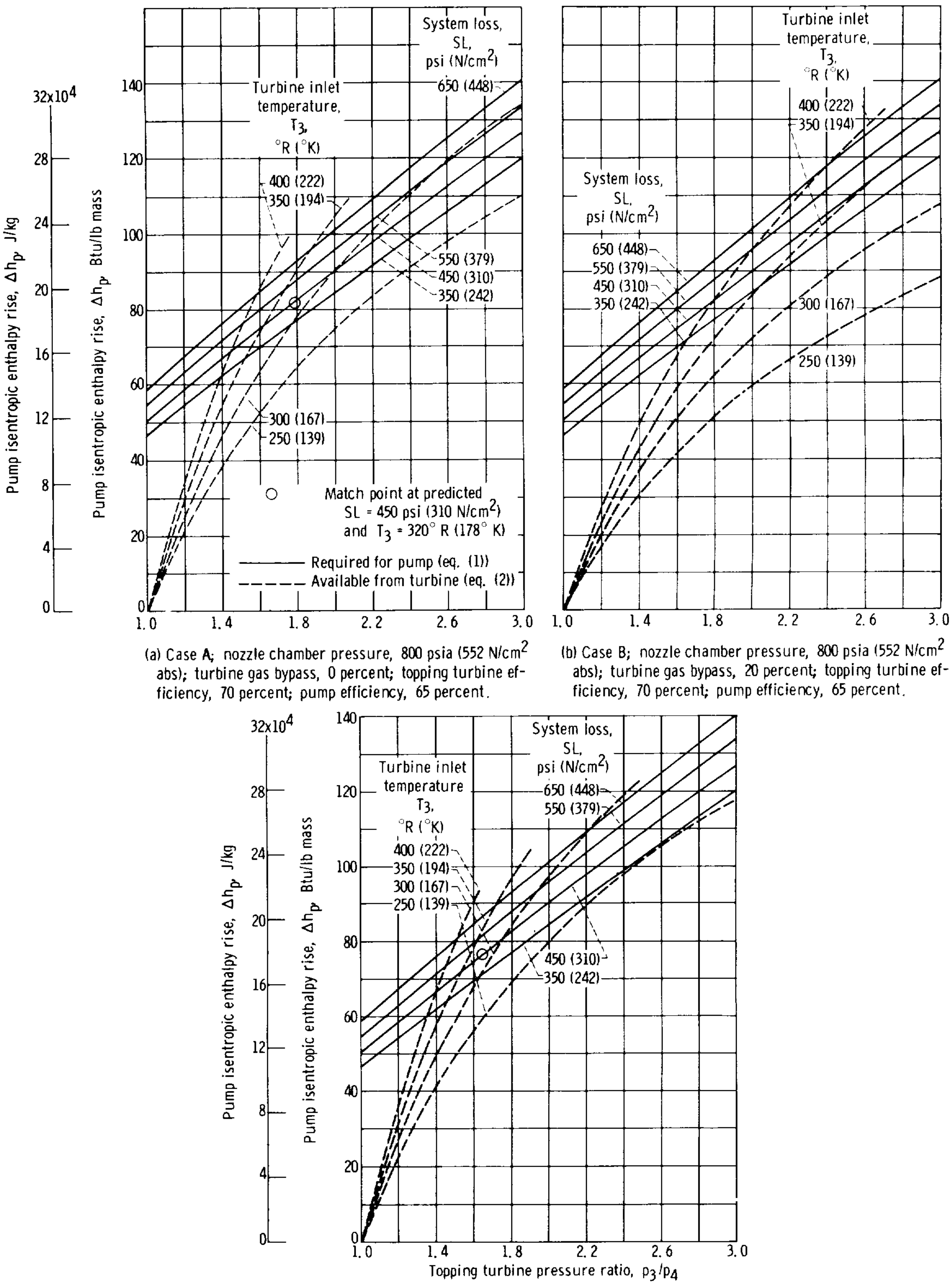
[redacted]



(a) Case A; nozzle chamber pressure, 800 psia (552 N/cm^2 abs); turbine gas bypass, 0 percent; topping turbine efficiency, 70 percent; pump efficiency, 65 percent.

(b) Case B; nozzle chamber pressure, 800 psia (552 N/cm^2 abs); turbine gas bypass, 20 percent; topping turbine efficiency, 70 percent; pump efficiency, 65 percent.

(c) Case C; nozzle chamber pressure, 800 psia (552 N/cm^2 abs); turbine gas bypass, 20 percent; topping turbine efficiency, 80 percent; pump efficiency, 76 percent.

Figure 3. - Topping turbine and pump work matching curves.

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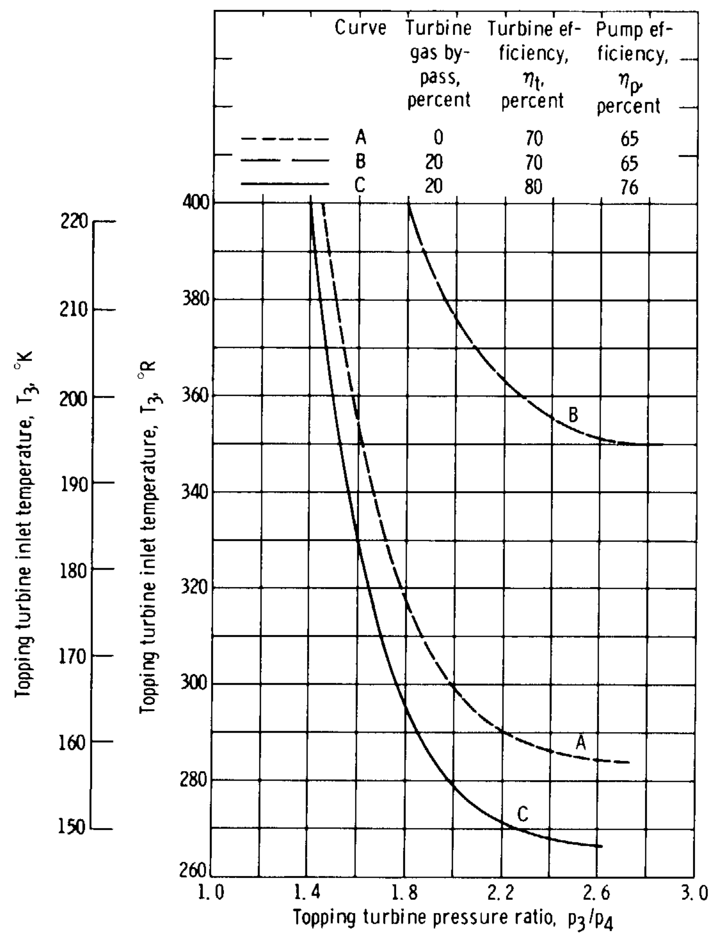

Figure 4. - Effect of topping turbine inlet temperature on turbine pressure ratio at matched conditions. Nozzle chamber pressure, 800 psia (552 N/cm^2 abs); system loss, 450 psi (310 N/cm^2).

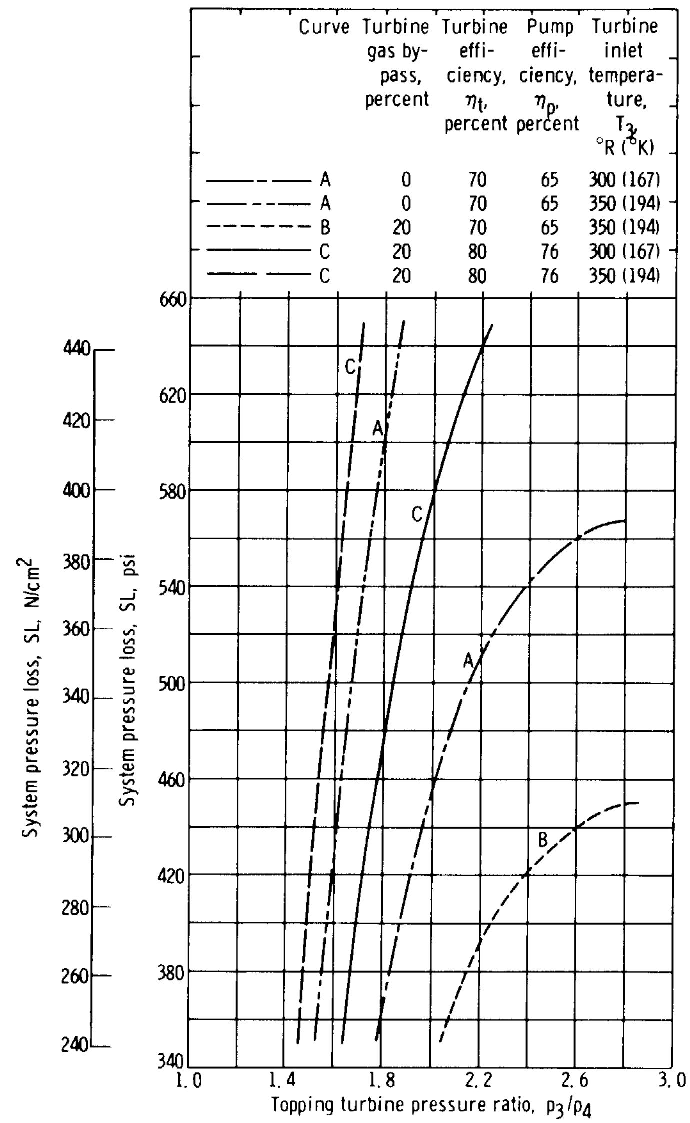

Figure 5. - Effect of system pressure loss on topping turbine pressure ratio at matched conditions. Nozzle chamber pressure, 800 psia (552 N/cm^2 abs).

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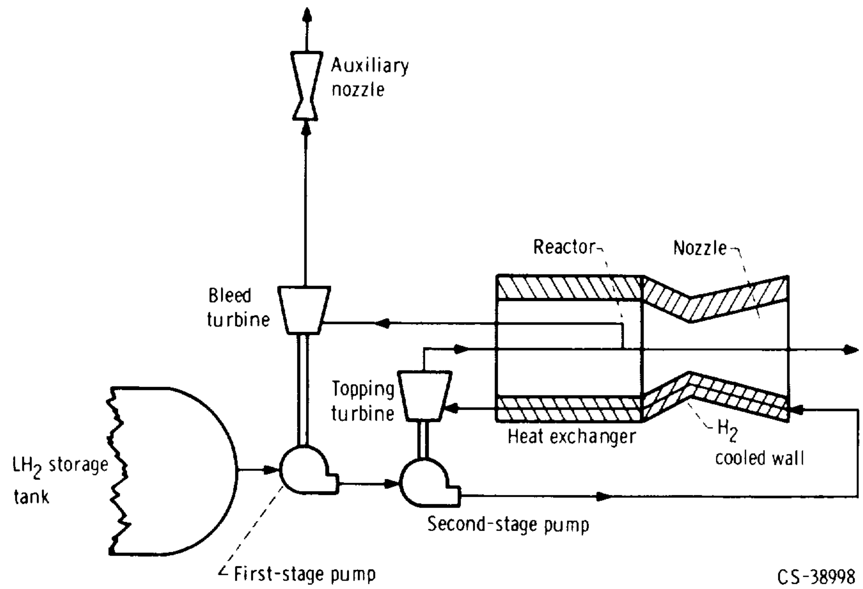

Figure 6. - Schematic drawing of split system.

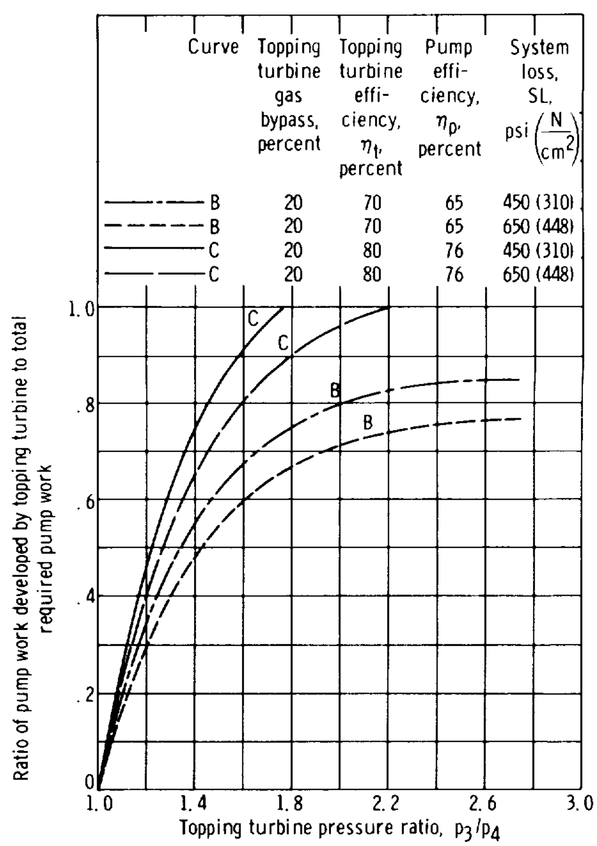

Figure 7. - Effect of bleed system and topping system work split on pump discharge pressure for split system. Nozzle chamber pressure, 800 psia (552 N/cm^2 abs); topping turbine inlet temperature, 300° R (167° K).

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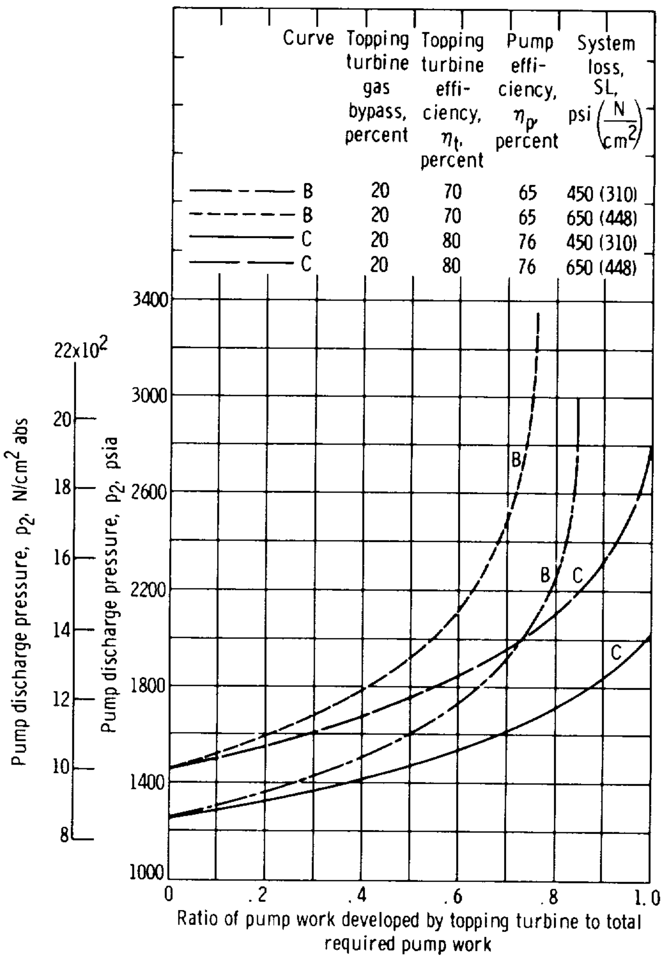
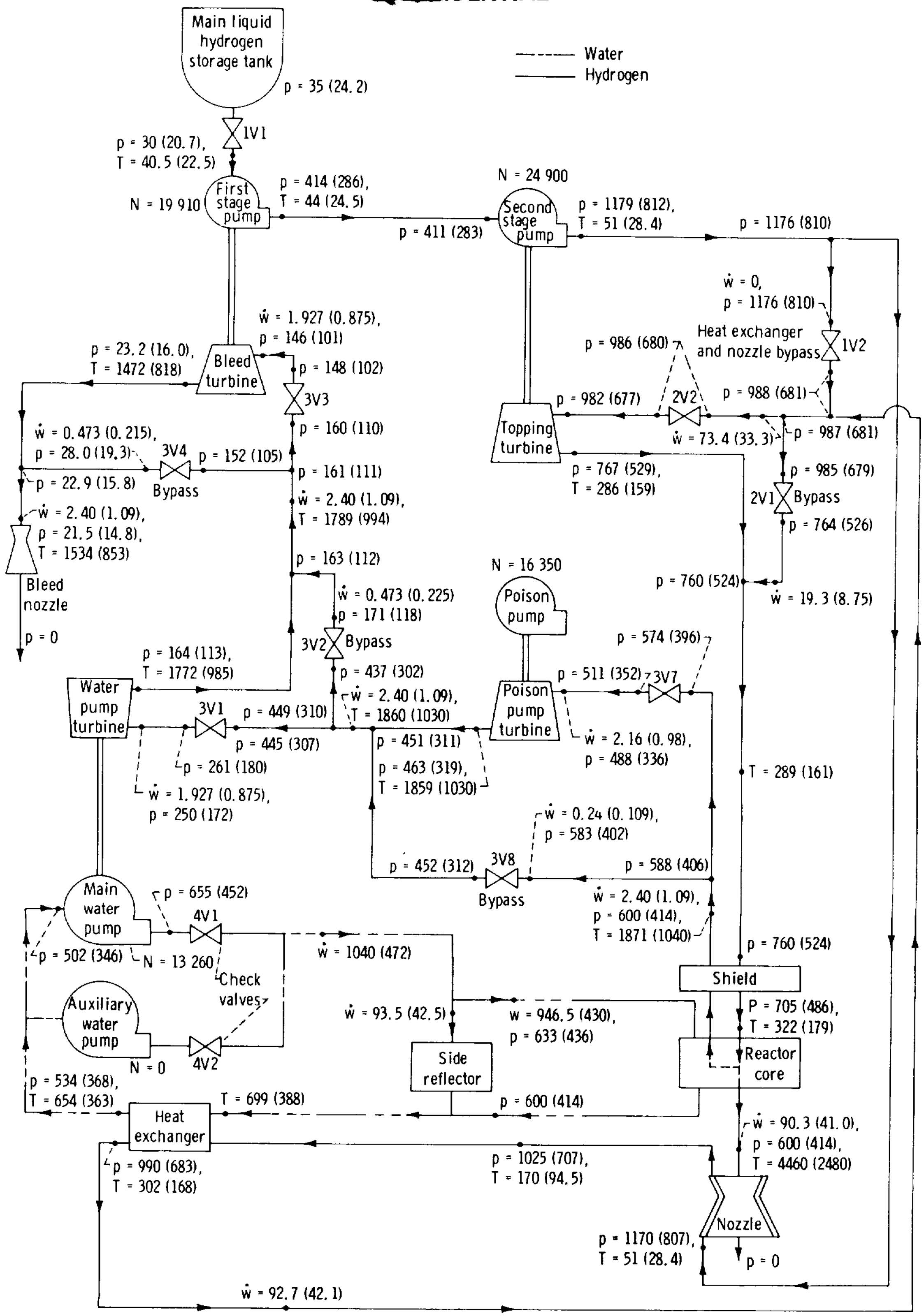
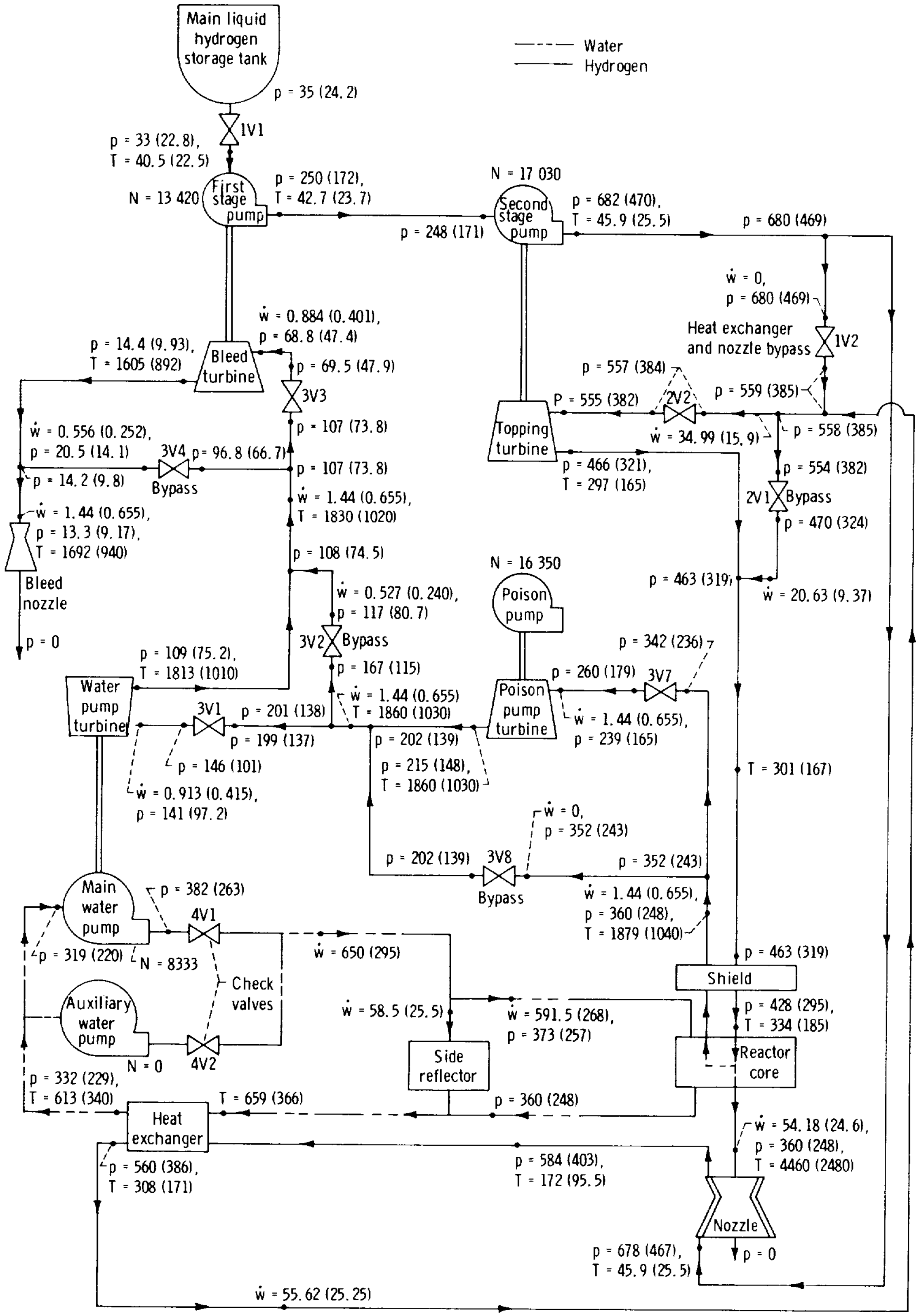


Figure 8. - Effect of bleed system and topping system work split on pump discharge pressure for split system. Nozzle chamber pressure, 800 psia (552 N/cm² abs); topping turbine inlet temperature, 300° R (167° K).



(a) Power, 100 percent.

Figure 9. - Flow schematic for 1500-megawatt TWMR reference system. Rotational speed, N, rpm; flow rate, ẇ, pound mass per second (kg/sec); absolute pressure, p, psia (N/cm^2 abs); temperature, T, °R (°K).



(b) Power, 60 percent.

Figure 9. - Continued.

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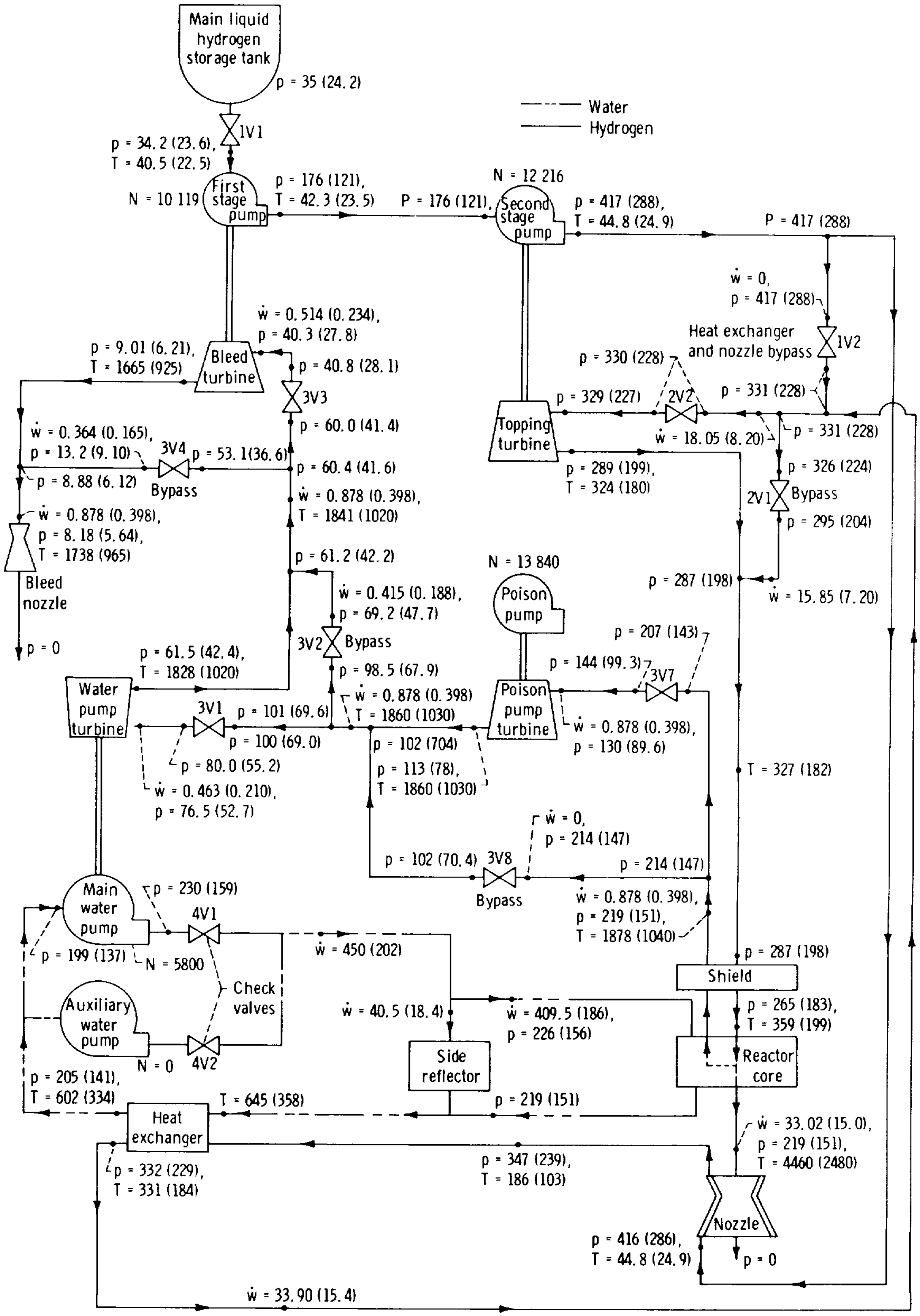

(c) Power, 37 percent.

Figure 9. - Concluded.

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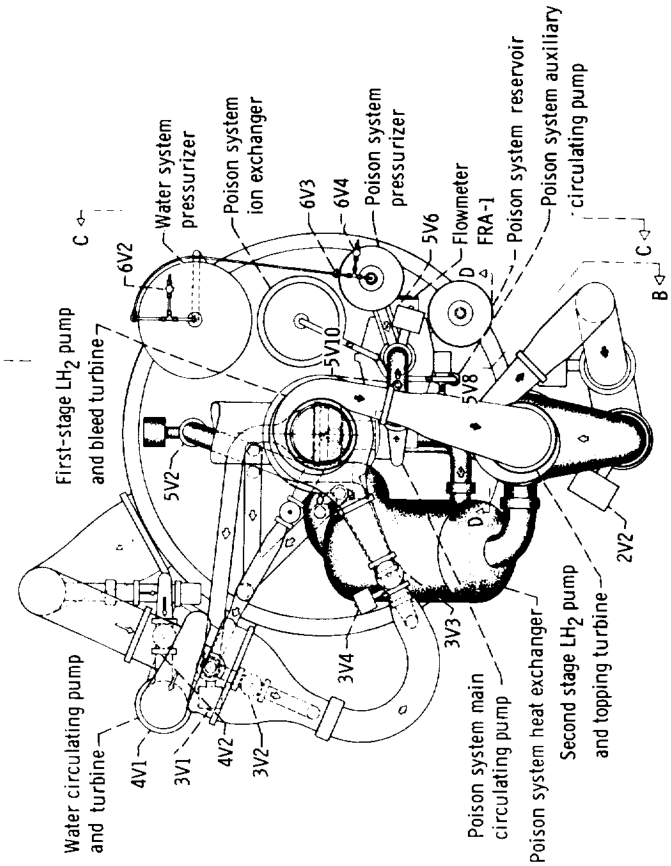
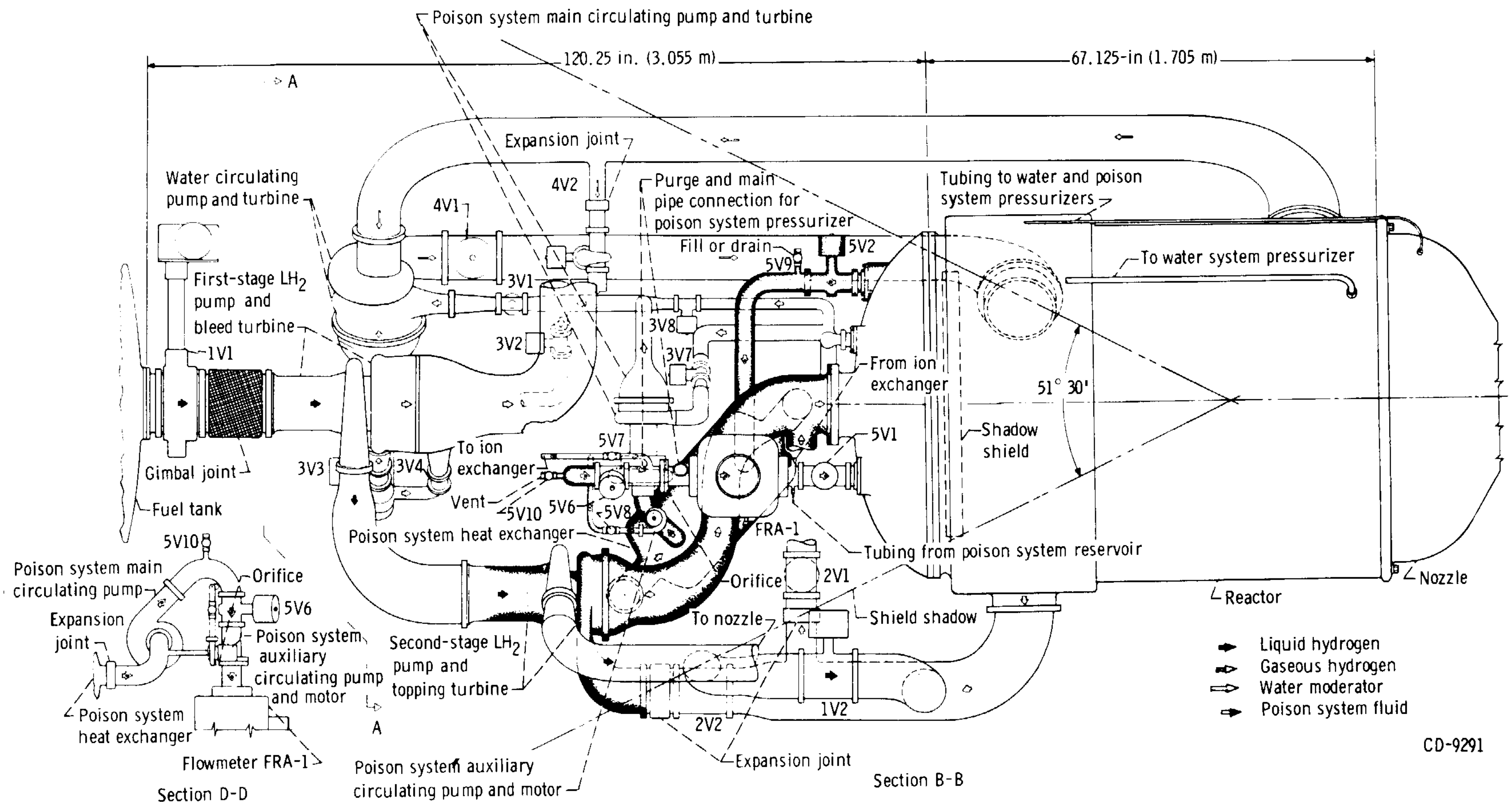
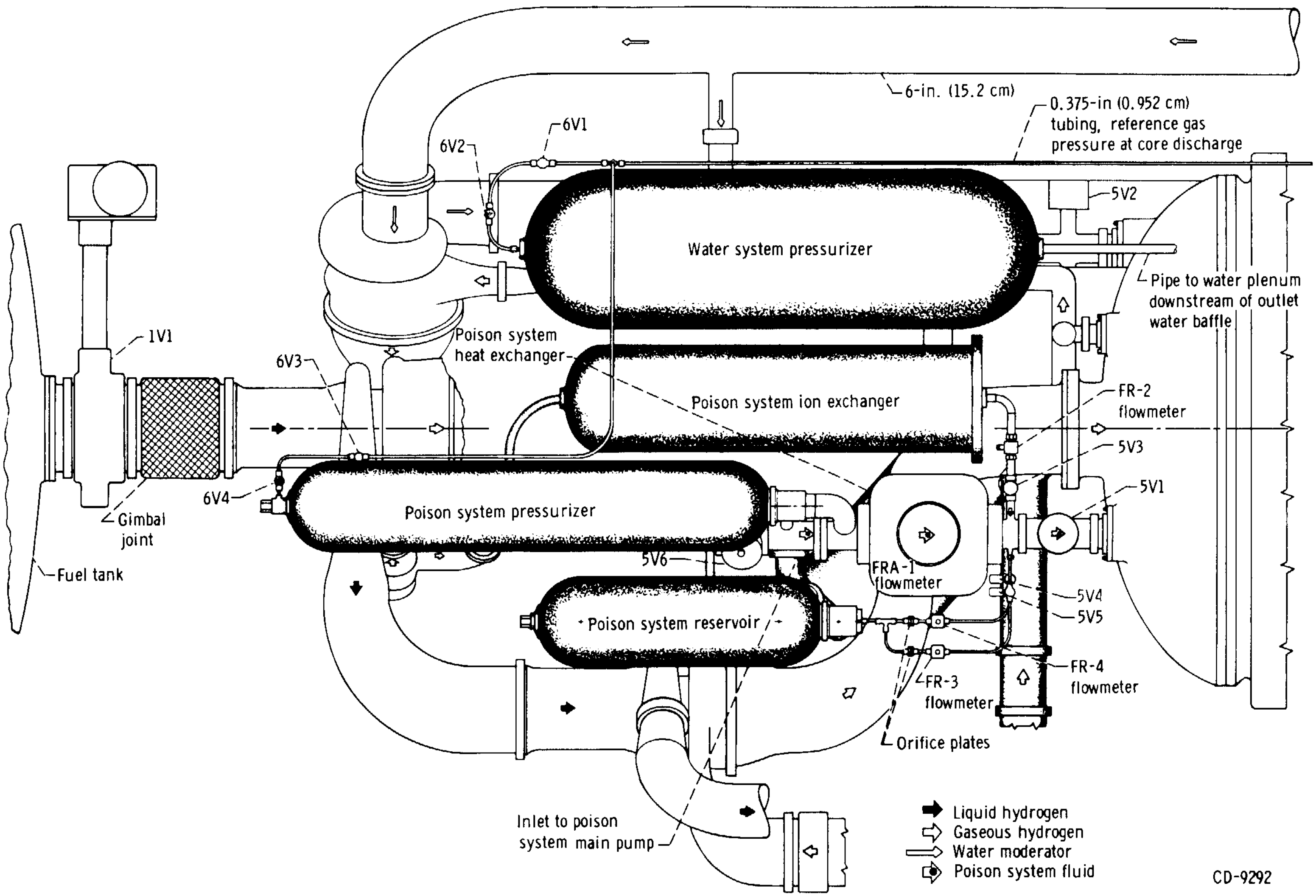

B
C
Water system pressurizer
Poison system ion exchanger
6V2
6V3
6V4
Poison system pressurizer
5V6
Flowmeter FRA-1
D
Poison system reservoir
Poison system auxiliary circulating pump
C
B
First-stage LH_2 pump and bleed turbine
5V10
5V8
5V2
D
2V2
Second stage LH_2 pump and topping turbine
Poison system heat exchanger
Poison system main circulating pump
3V3
3V4
Water circulating pump and turbine
4V1
3V1
4V2
3V2

Section A-A



(a) Detail of sections A-A, B-B, and D-D.

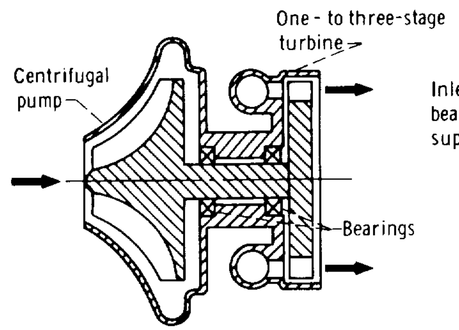
Figure 10. - Arrangement of external system components of TWMR.



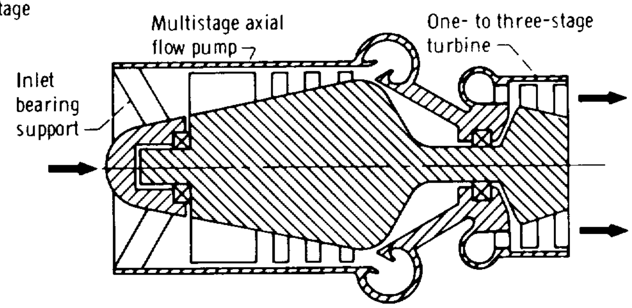
Section C-C

(b) Detail of section C-C.

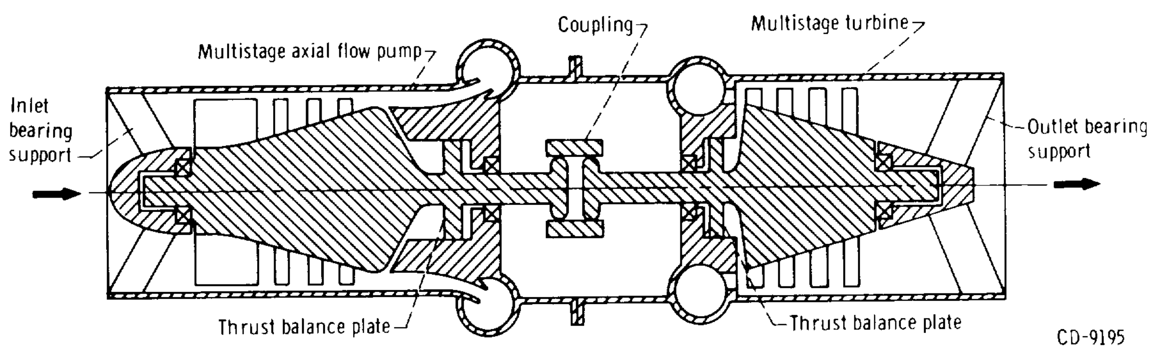
Figure 10. - Concluded.



(a) Midbearing support (overhung pump and turbine).

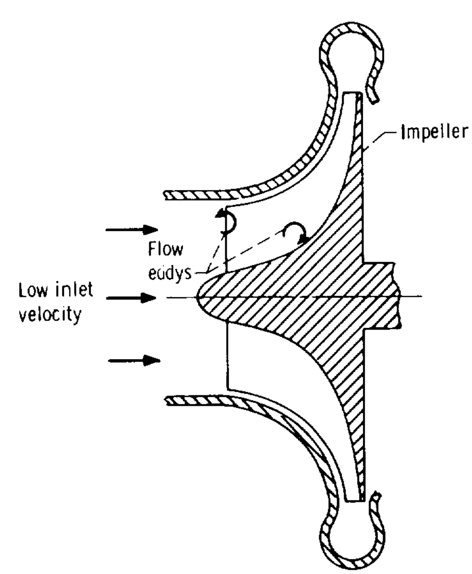


(b) Straddle mounted pump and overhung turbine.

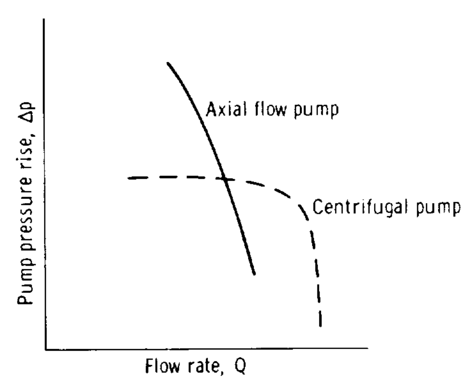


(c) Straddle mounted pump and turbine.

Figure 11. - Pumping machinery types and supports.



(a) Centrifugal pump designed for high suction specific speed.



(b) Pump performance characteristic.

Figure 12. - Pump stability.

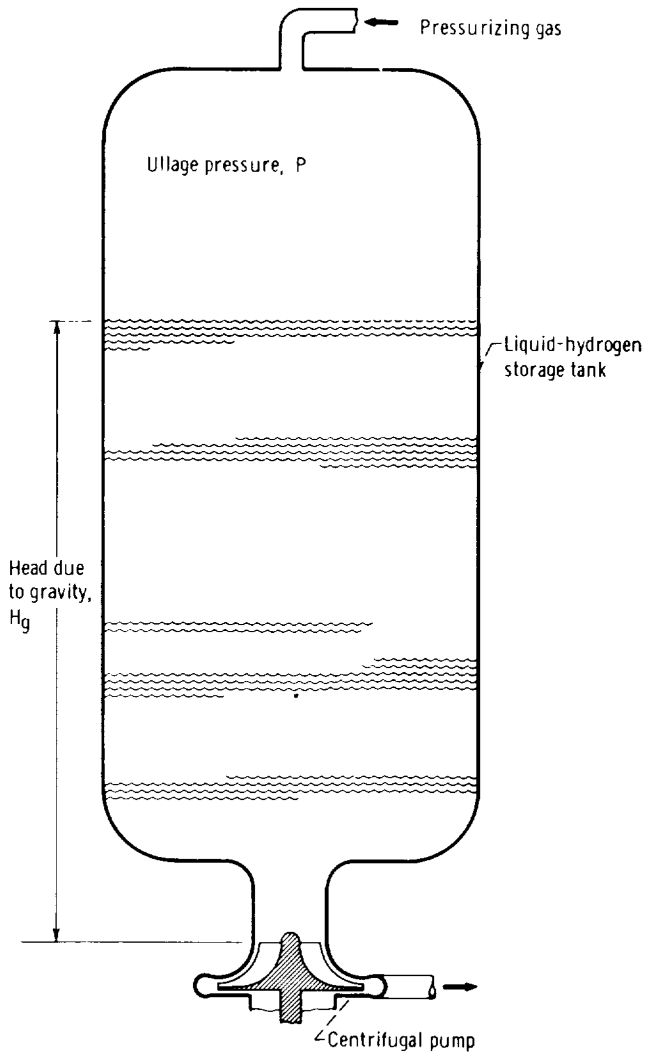
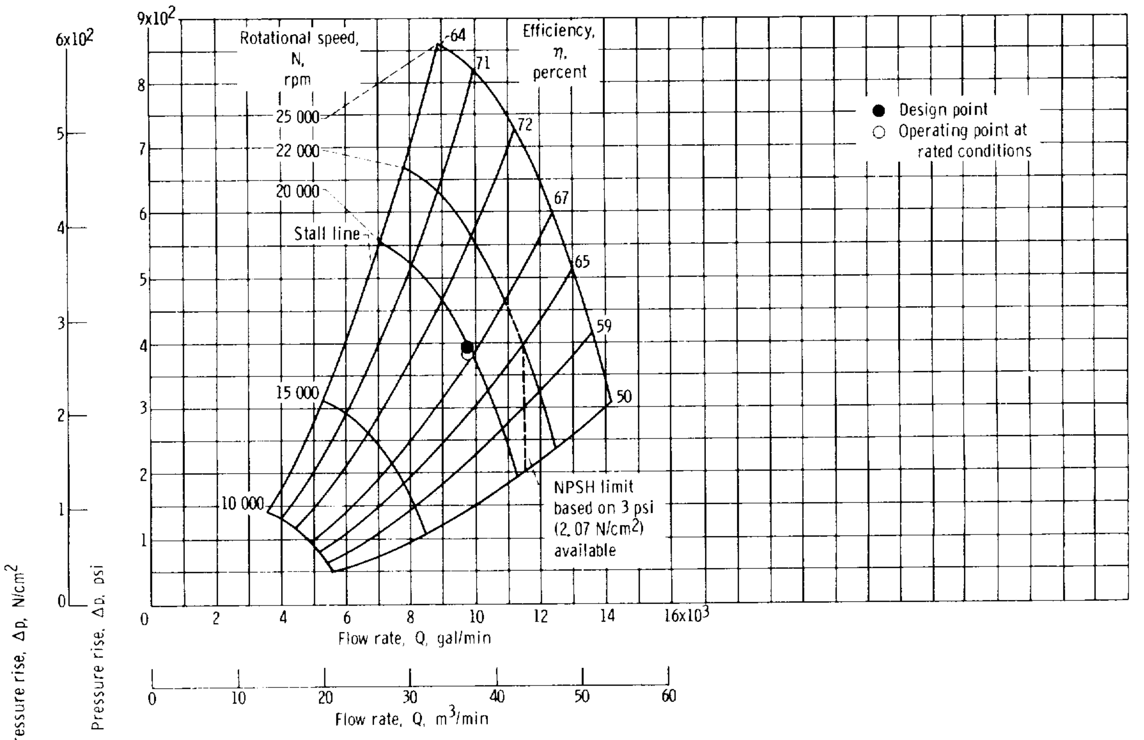
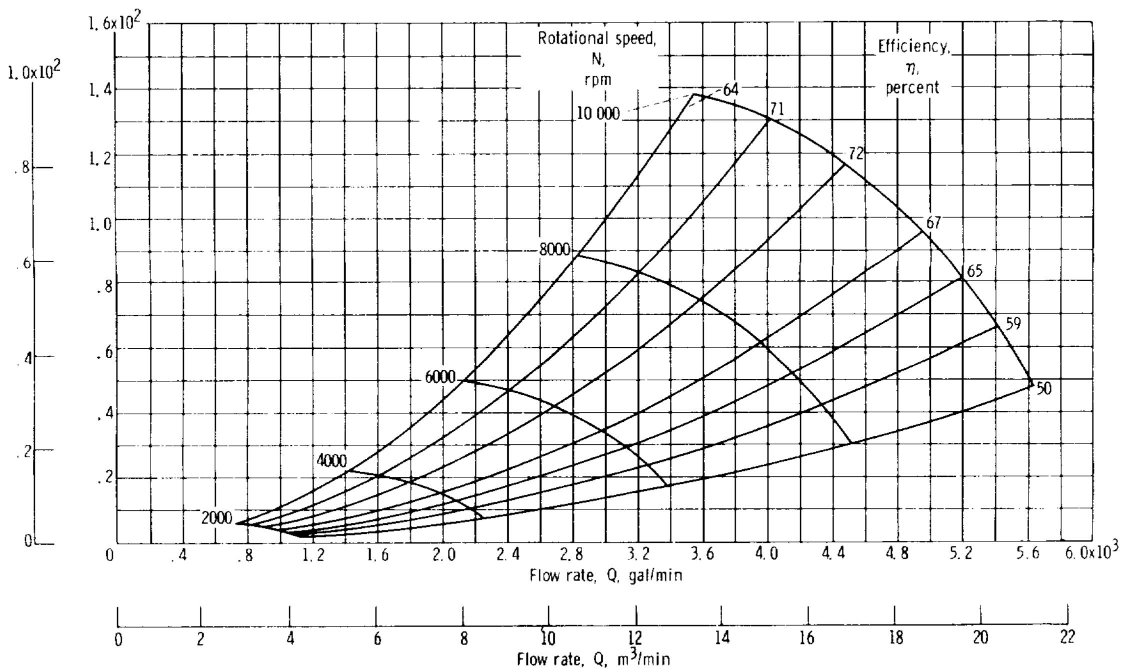


Figure 13. - Ullage pressure and gravity head.

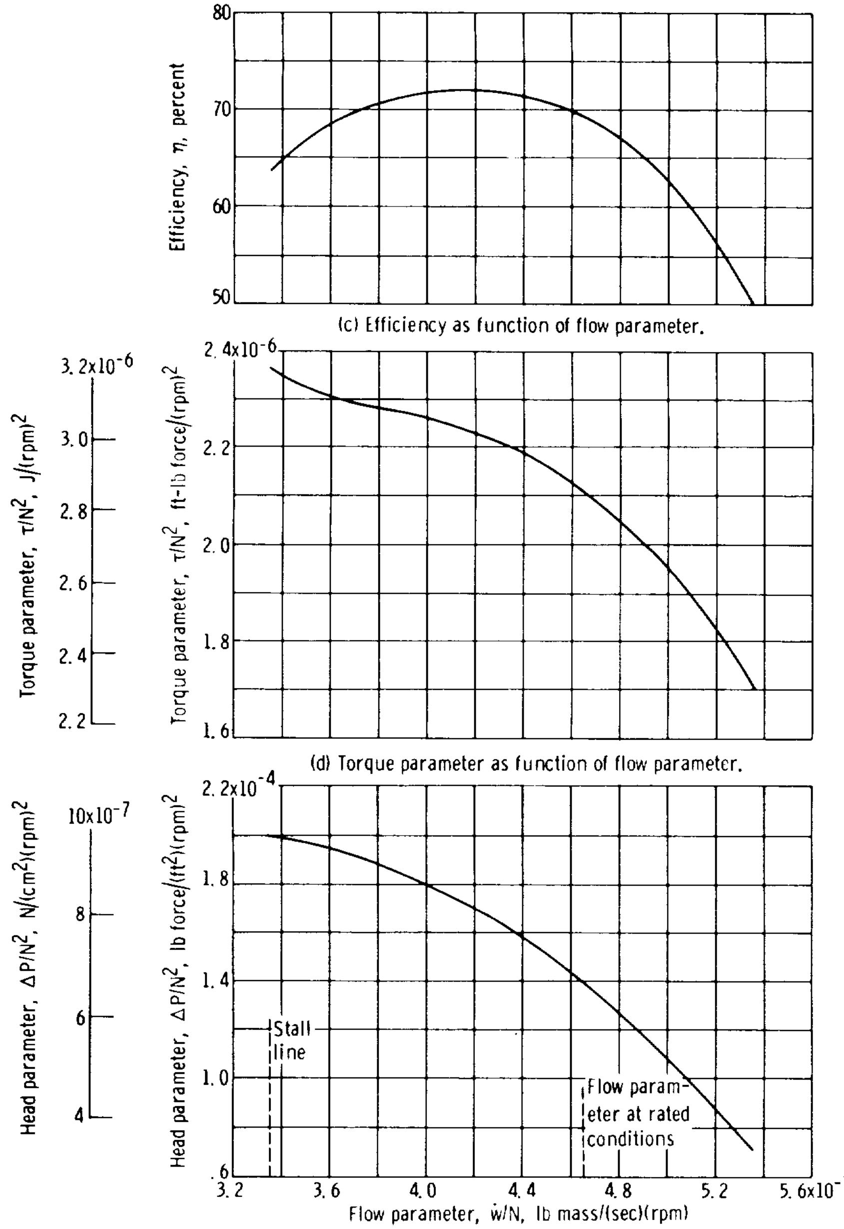


(a) Pressure rise as function of flow rate for high speeds.



(b) Pressure rise as function of flow rate for low speeds.

Figure 14. - First-stage liquid-hydrogen (bleed) pump performance. Flow rate, 92.7 pounds mass per second (42.1 kg/sec) or 9789 gallons per minute (37.0 m^3/min); hydrogen density, 42.5 pounds mass per cubic foot (68.1 kg/m^3). Design point conditions: pressure rise, 395 psi (272 N/cm^2); rotational speed, 20 000 rpm. Operating point at rated conditions: pressure rise, 384.3 psi (265 N/cm^2); rotational speed, 19 910 rpm.

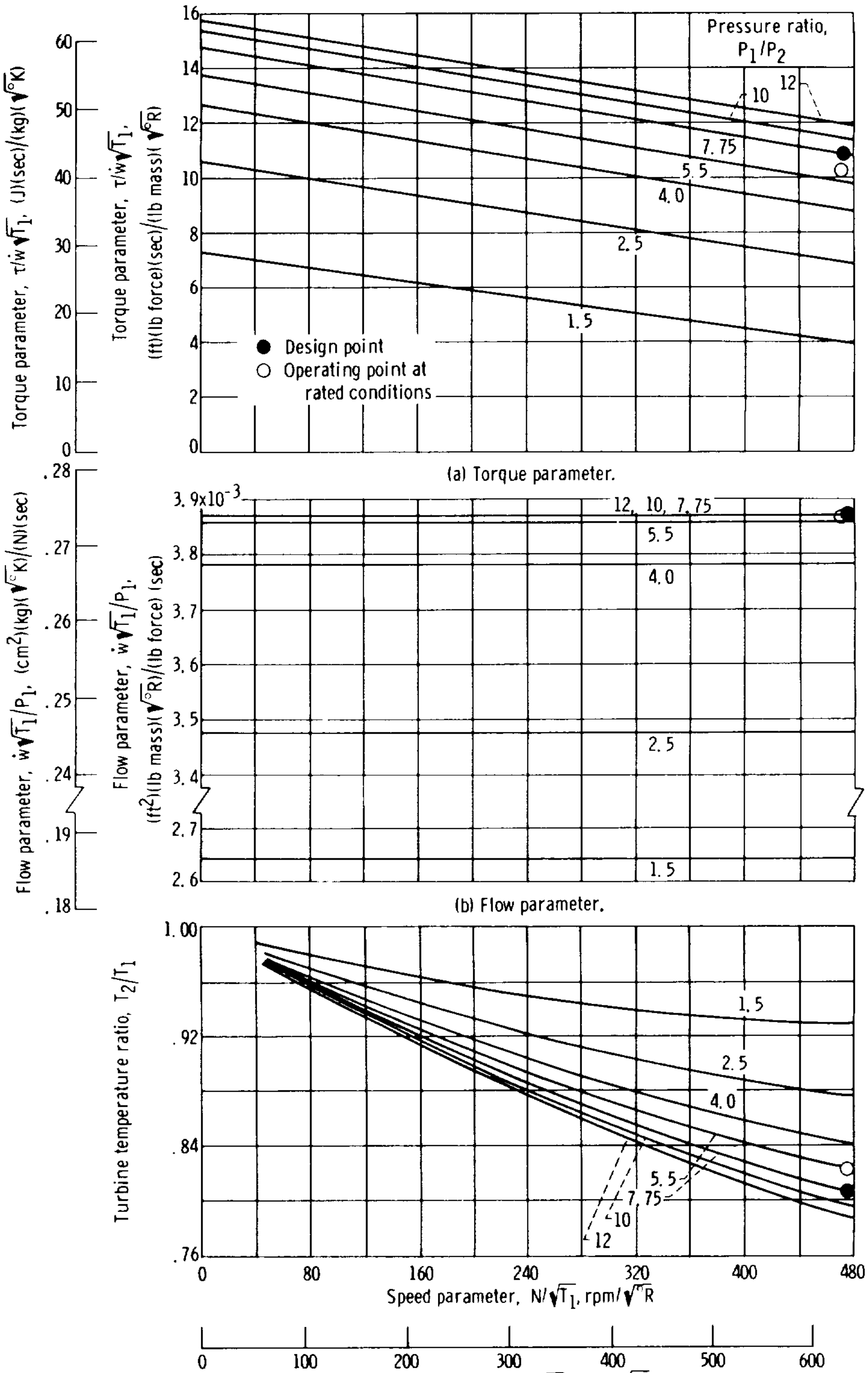


(c) Efficiency as function of flow parameter.

(d) Torque parameter as function of flow parameter.

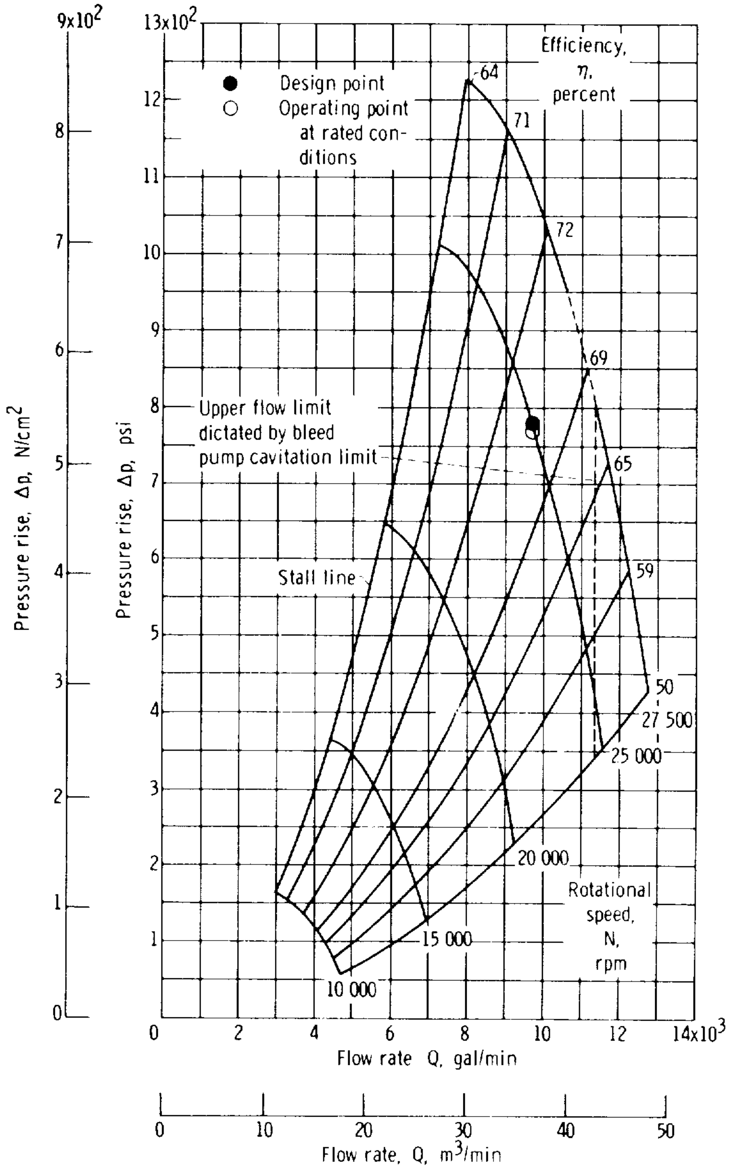
(e) Head parameter as function of flow parameter.

Figure 14. - Concluded.



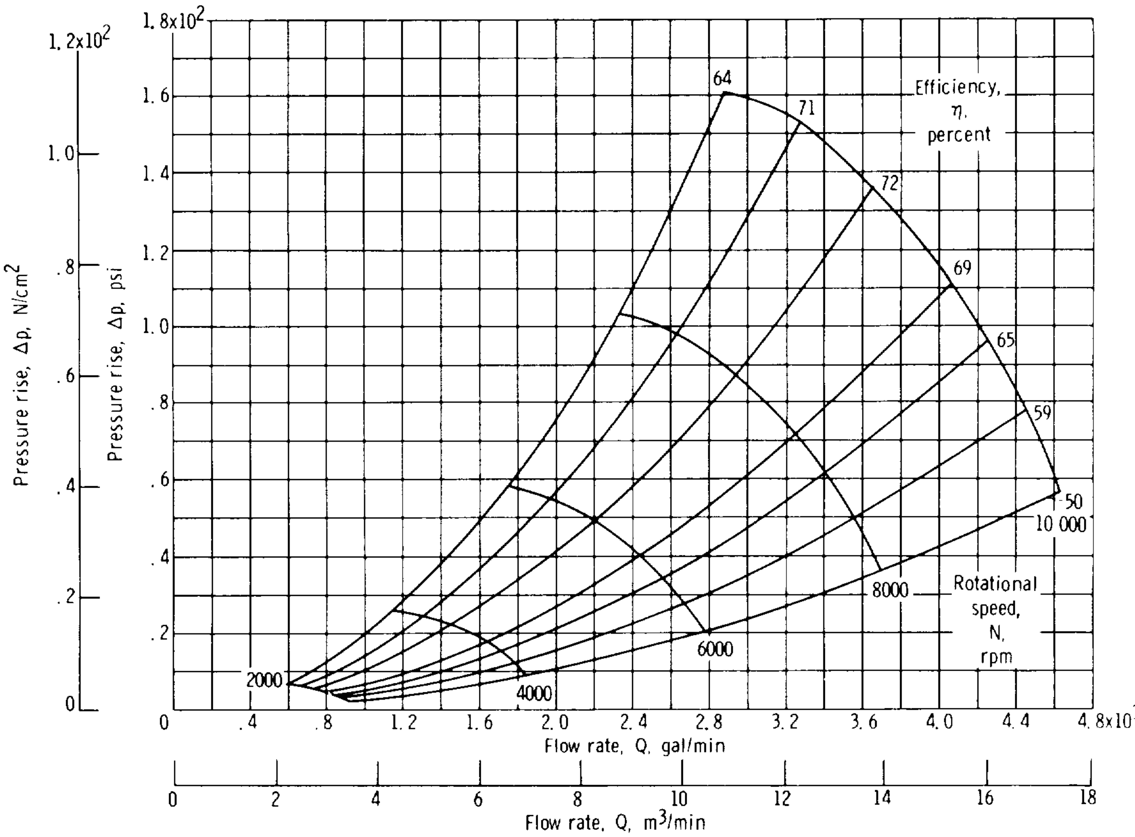
(c) Temperature ratio.

Figure 15. - Bleed turbine performance as function of speed parameter. Design conditions: rotational speed, 20 000 rpm; power, 3368 horsepower (2.51 MW); inlet temperature, 1781° R (990° K); outlet temperature, 1435° R (797° K); inlet pressure, 145.9 psia (101 N/cm^2 abs); outlet pressure, 19.2 psia (13.2 N/cm^2 abs); flow rate, 1.927 pounds mass per second (0.875 kg/sec).



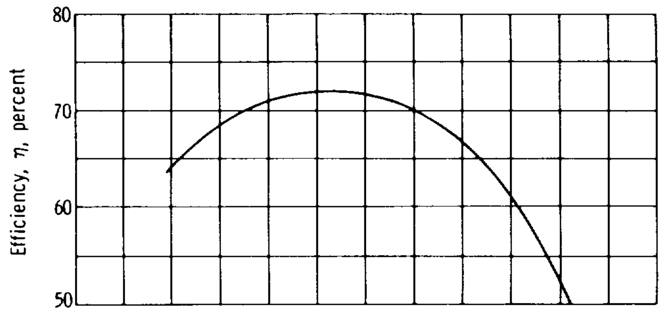
(a) Pressure rise as function of flow rate for high speeds.

Figure 16. - Second-stage liquid-hydrogen (topping) pump performance. Flow rate, 92.7 pounds mass per second (42.1 kg/sec) or 9675 gallons per minute (36.6 m^3/min); hydrogen density, 4.3 pounds mass per cubic foot (68.7 kg/m^3). Design point conditions: pressure rise, 778 psi (536 N/cm^2); rotational speed, 25 000 rpm. Operating point at rated conditions: pressure rise, 768.5 psi (530 N/cm^2); rotation speed, 24 900 rpm.

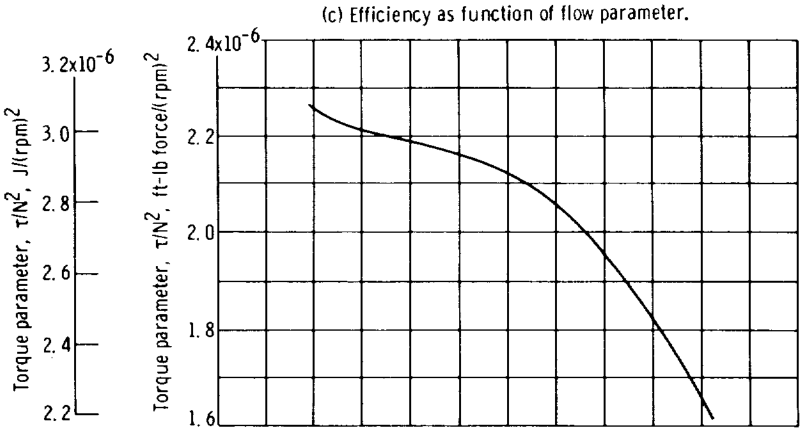


(b) Pressure rise as function of flow rate for low speeds.

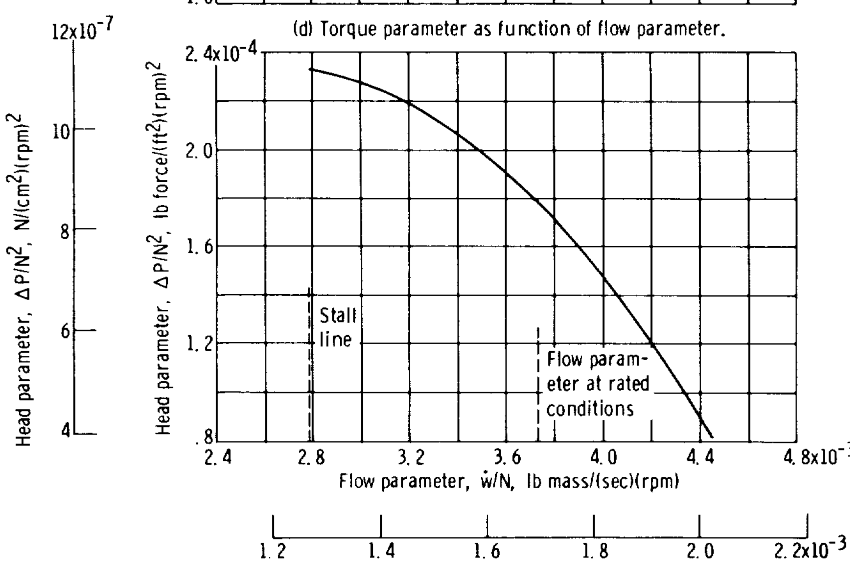
Figure 16. - Continued.



(c) Efficiency as function of flow parameter.



(d) Torque parameter as function of flow parameter.



(e) Head parameter as function of flow parameter.

Figure 16. - Concluded.

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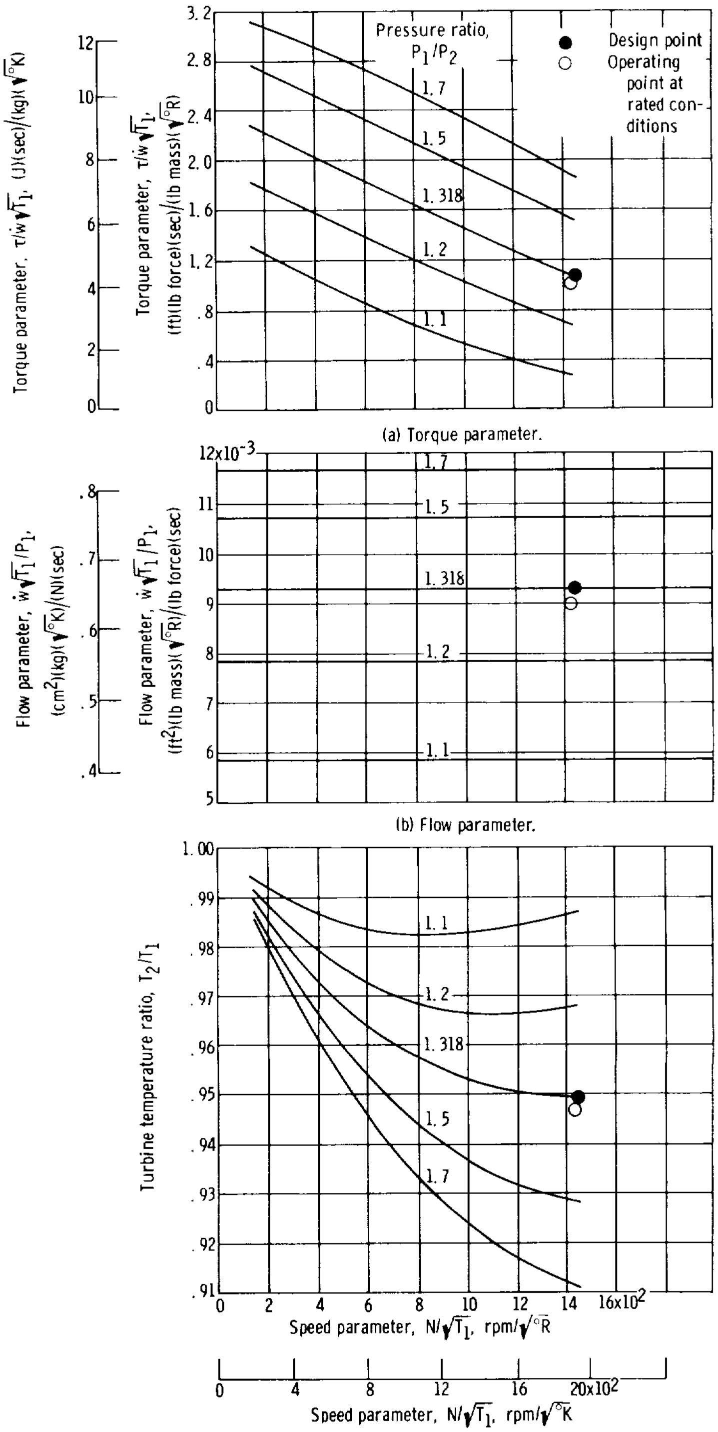

Figure 17. - Topping turbine performance as function of speed parameter. Design conditions: rotational speed, 25 000 rpm; power, 6516.3 horsepower (4.86 MW); inlet temperature, 300° R (167° K); outlet temperature, 285° R(158° K); inlet pressure, 955 psia (659 N/cm² abs); outlet pressure, 725 psia (500 N/cm² abs); flow rate, 74.1 pounds mass per second (33.6 kg/sec).

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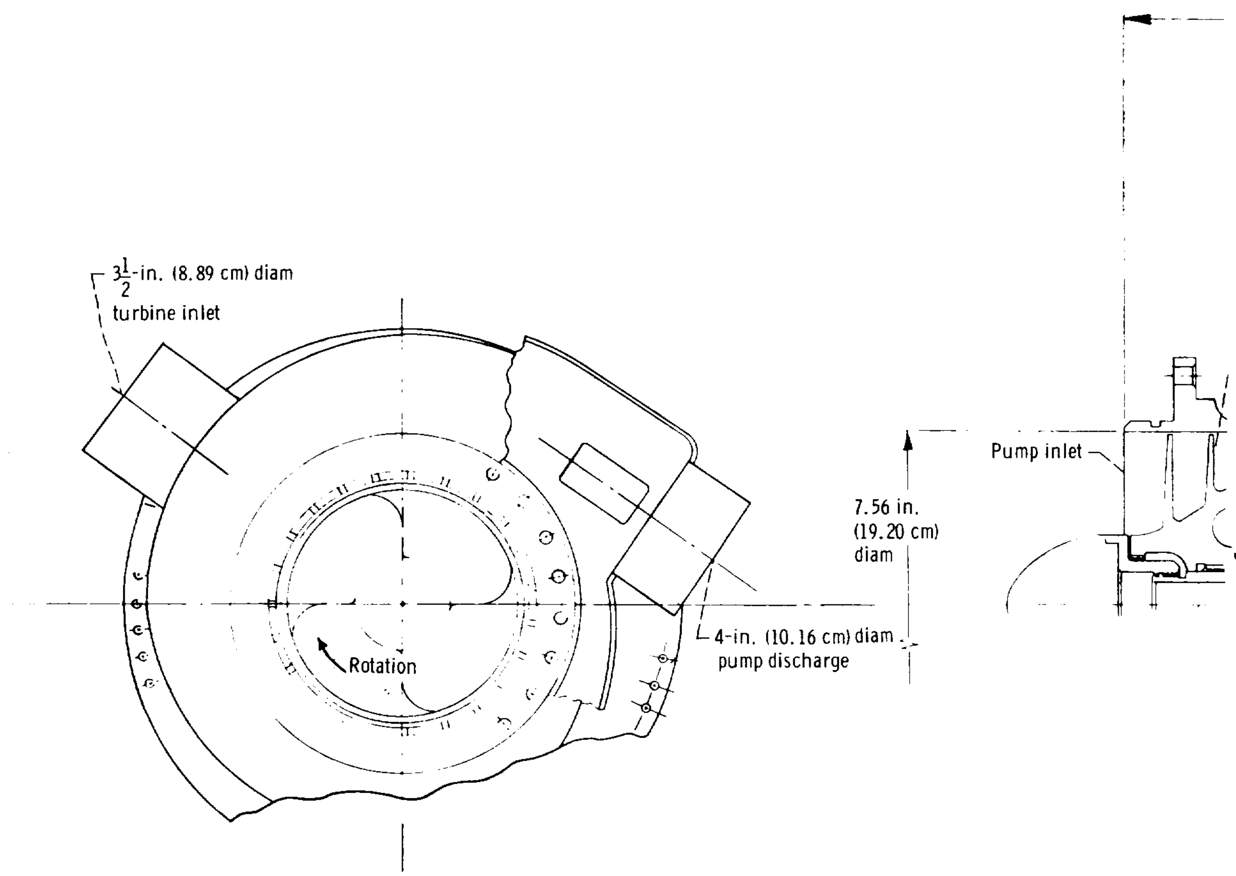
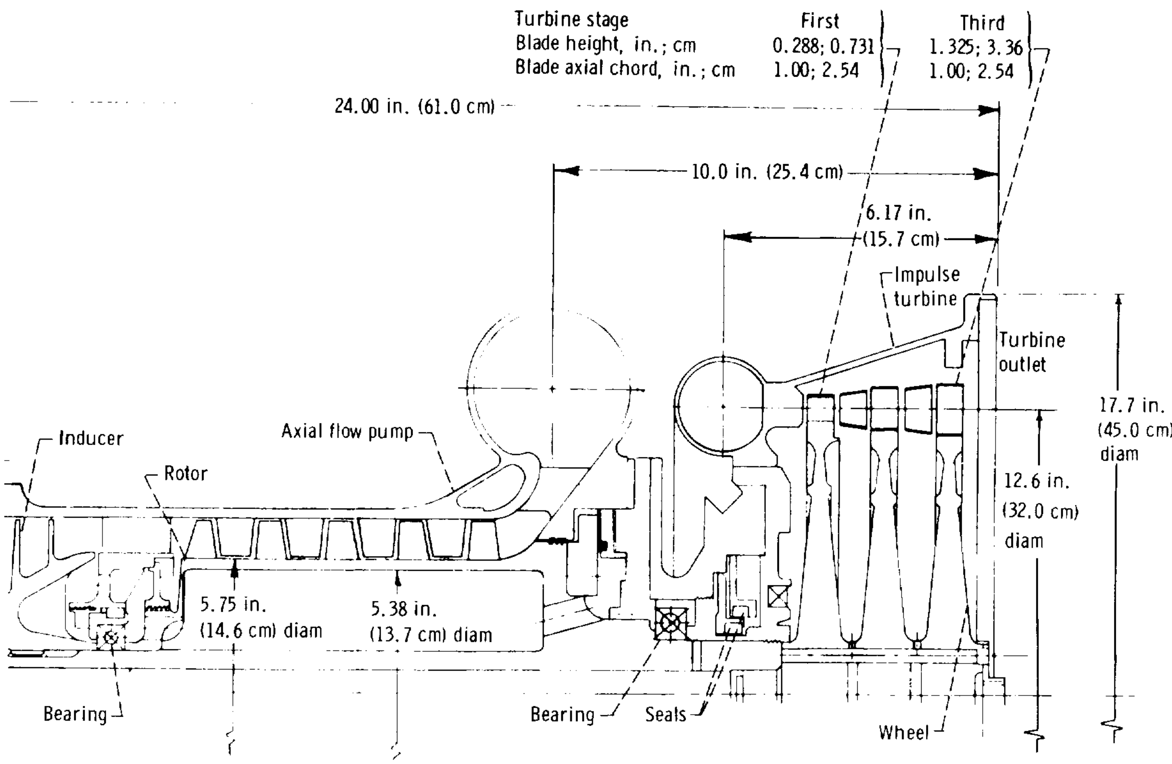


Figure 18. - Bleed turbopump mechanical design layout. Estimated weight, 349 pounds mass (15



(158 kg); moment of inertia of rotating assembly about axis, 2.774 inch pound force second squared (0.314 (kg)(m^2)).

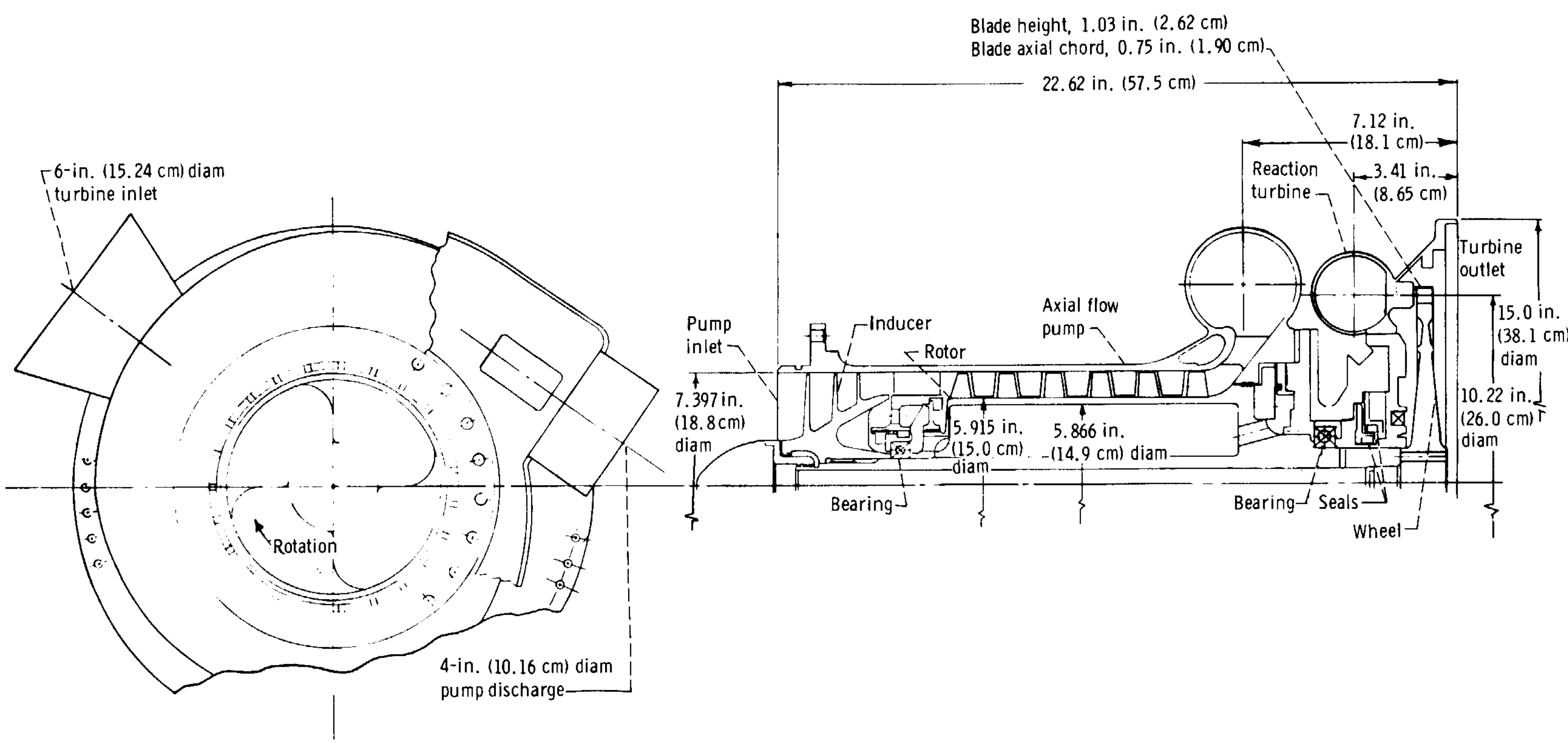
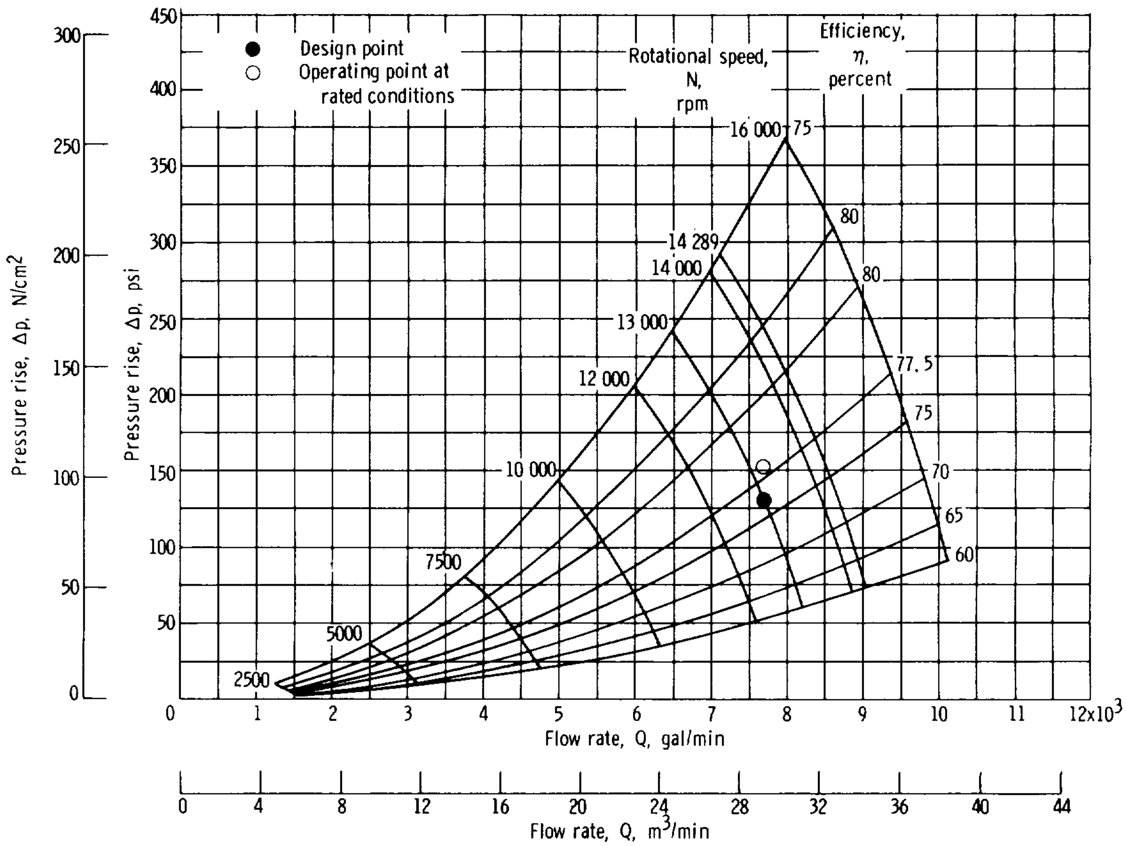
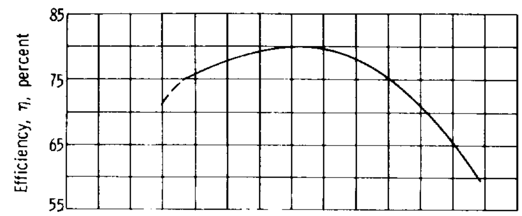


Figure 19. - Topping turbopump mechanical design layout. Estimated weight, 349 pounds mass (158 kg); moment of inertia of rotating assembly about axis, 2.774 inch pound force second squared (0.314 (kg)(m^2)).

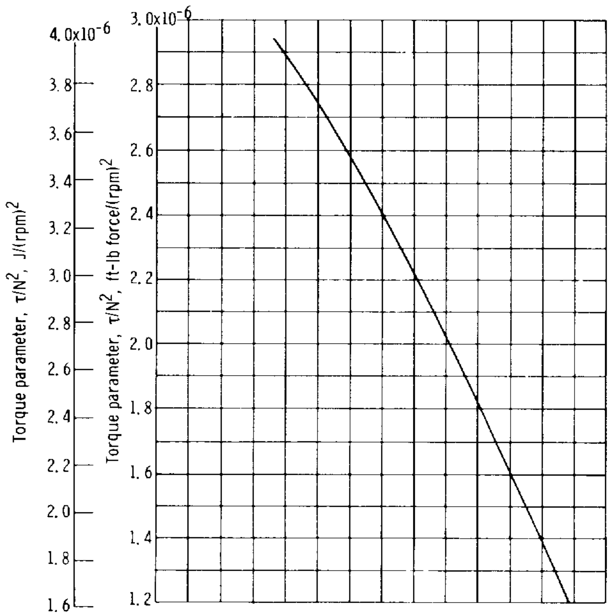


(a) Pressure rise as function of flow rate.

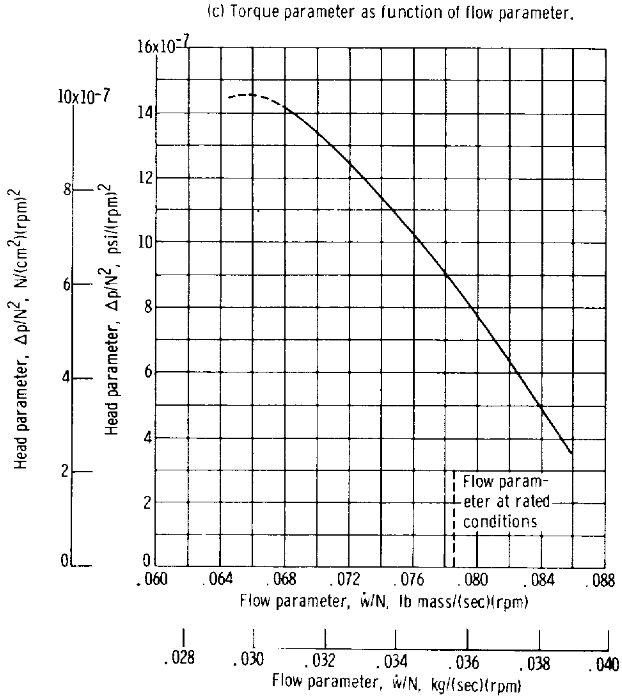
Figure 20. - Water pump performance. Water temperature, 644° R (358° K); water density, 60.6 pounds mass per cubic foot (970 kg/m³); flow rate, 1040 pounds mass per second (472 kg/sec) or 7733 gallons per minute (29.2 m³/min). Design point conditions: pressure rise, 131 psi (90.3 N/cm²); rotational speed, 12 990 rpm. Operating point at rated conditions: pressure rise, 153 psi (106 N/cm²); rotational speed, 13 254 rpm.



(b) Efficiency as function of flow parameter.

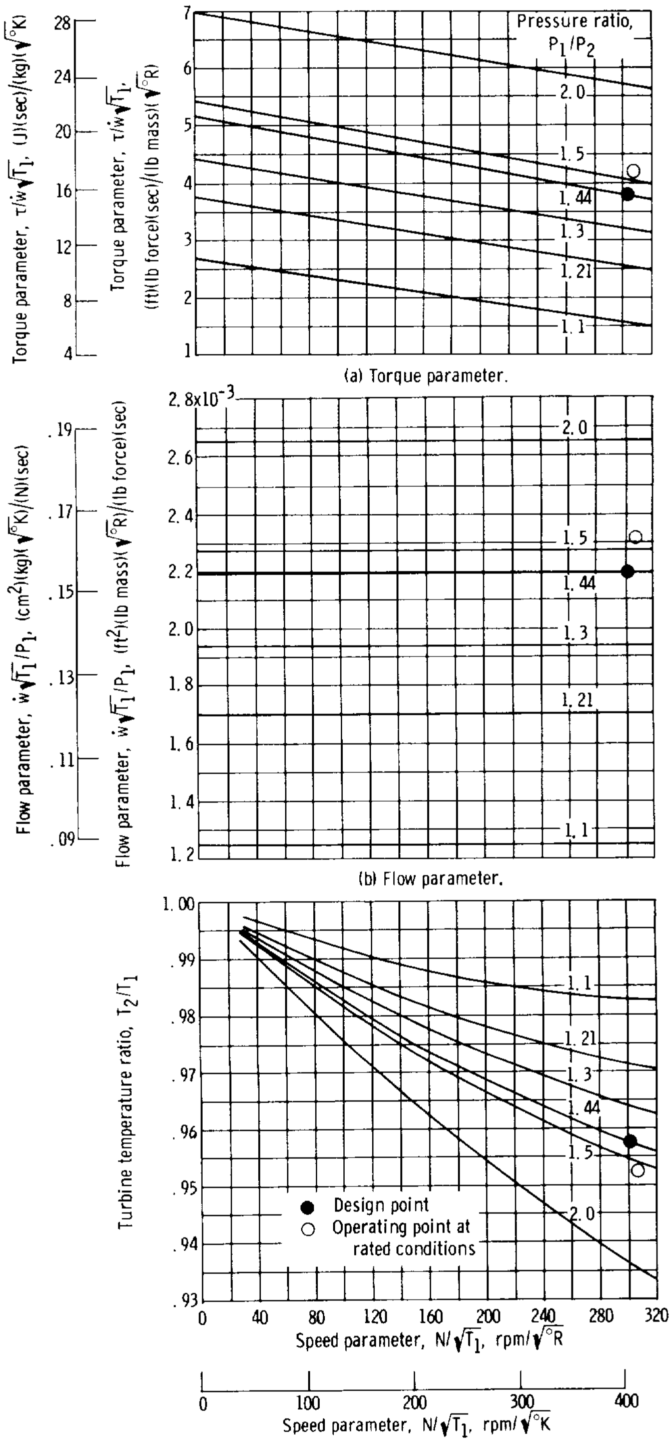


(c) Torque parameter as function of flow parameter.



(d) Head parameter as function of flow parameter.

Figure 20. - Concluded.



(a) Torque parameter.

(b) Flow parameter.

(c) Temperature ratio.

Figure 21. - Water pump turbine performance as function of speed parameter. Design conditions: rotational speed, 12 990 rpm; power, 782.7 horsepower (583 kW); inlet temperature, 1860° R (1030 °K); outlet temperature, 1781° R (990 °K; inlet pressure, 263 psia (181 N/cm² abs); flow rate, 1.927 pounds mass per second (0.875 kg/sec).

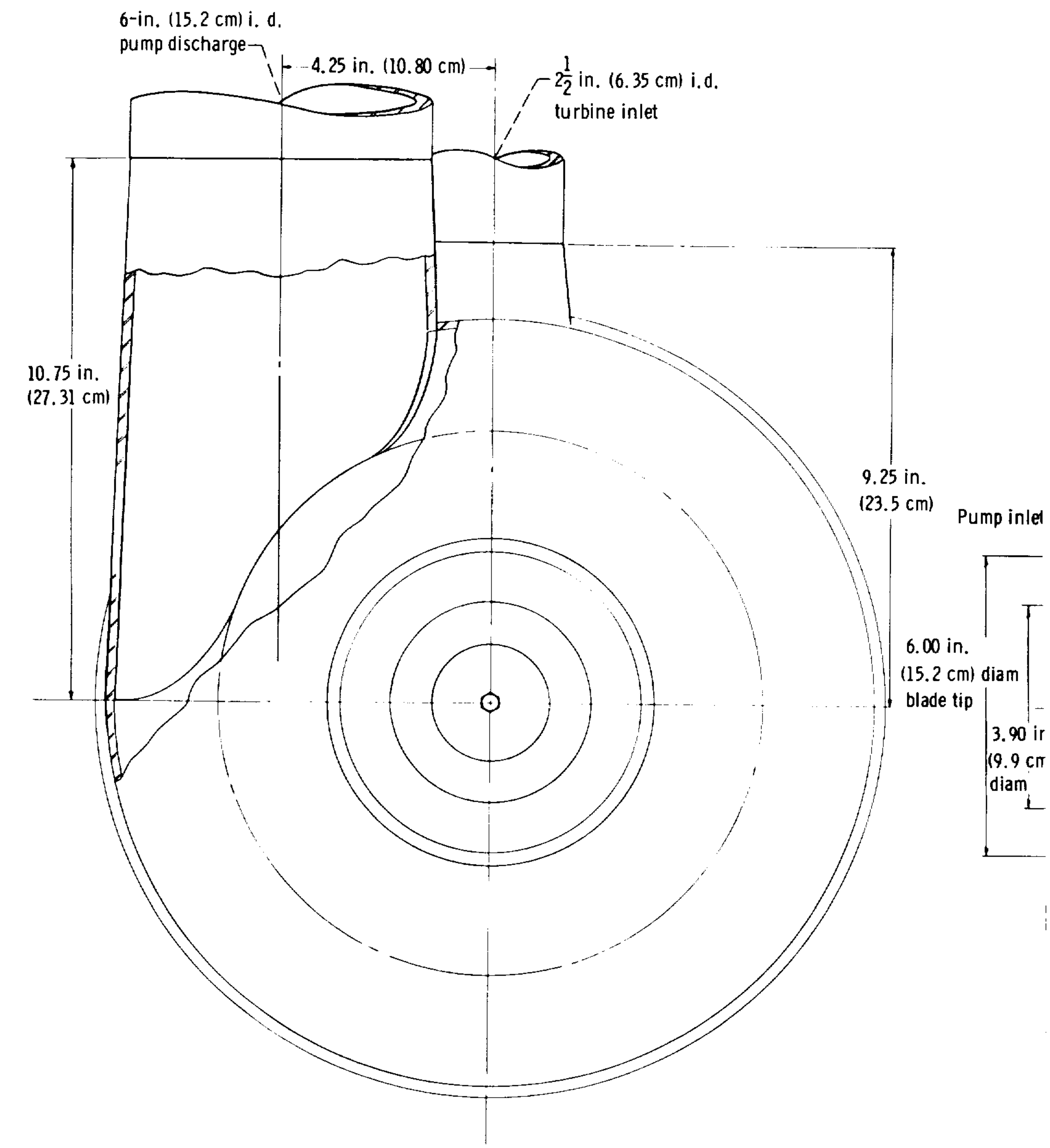
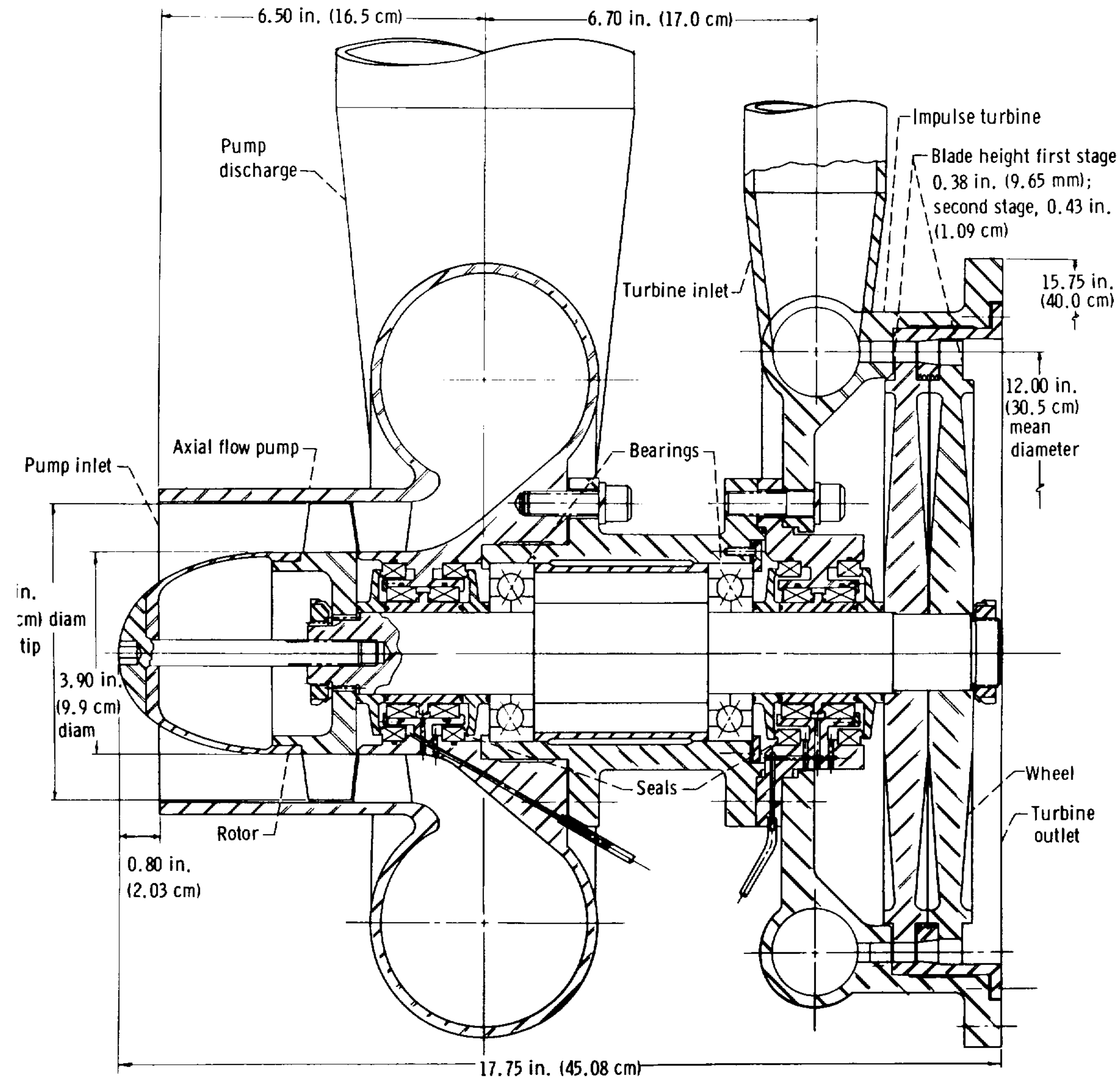


Figure 22. - Water turbopump mechanical design layout. Moment of inertia of rotating assembly about axis, 2.0 inch pou
320 cubic inches (5250 cm^3); turbine weight, 128 pounds mass (58.1 kg); pump weight, 88 pounds mass (40 kg); pump wett



0 inch pound force second squared (0.226 (kg)(m^2)); total rotating weight, 60 pounds mass (27.2 kg); turbine fluid volume,
; pump wetted surface area, 970 square inches (6260 cm^2); pump fluid volume, 660 cubic inches (10 800 cm^3).

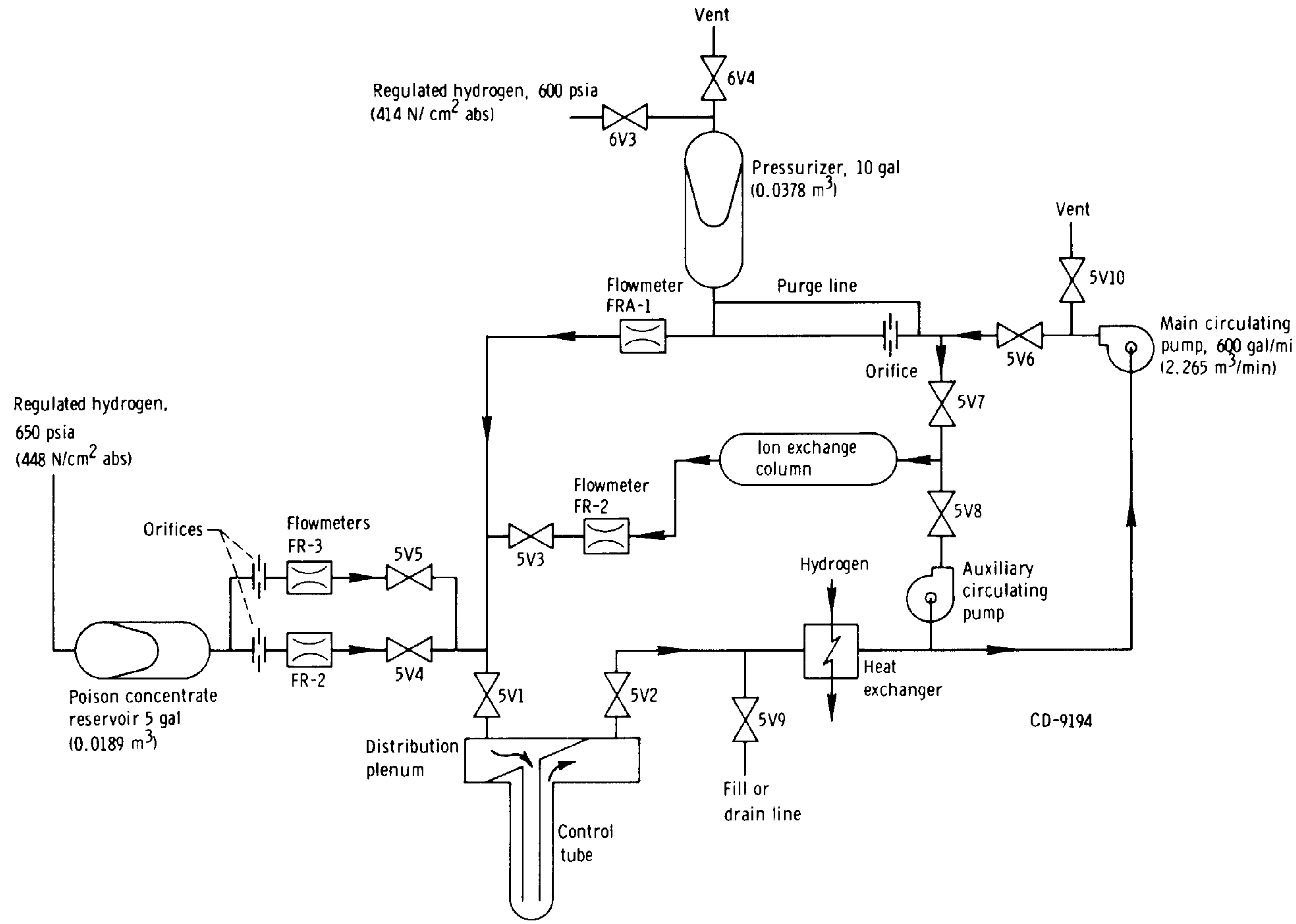


Figure 23. - Chemical poison system flow diagram.

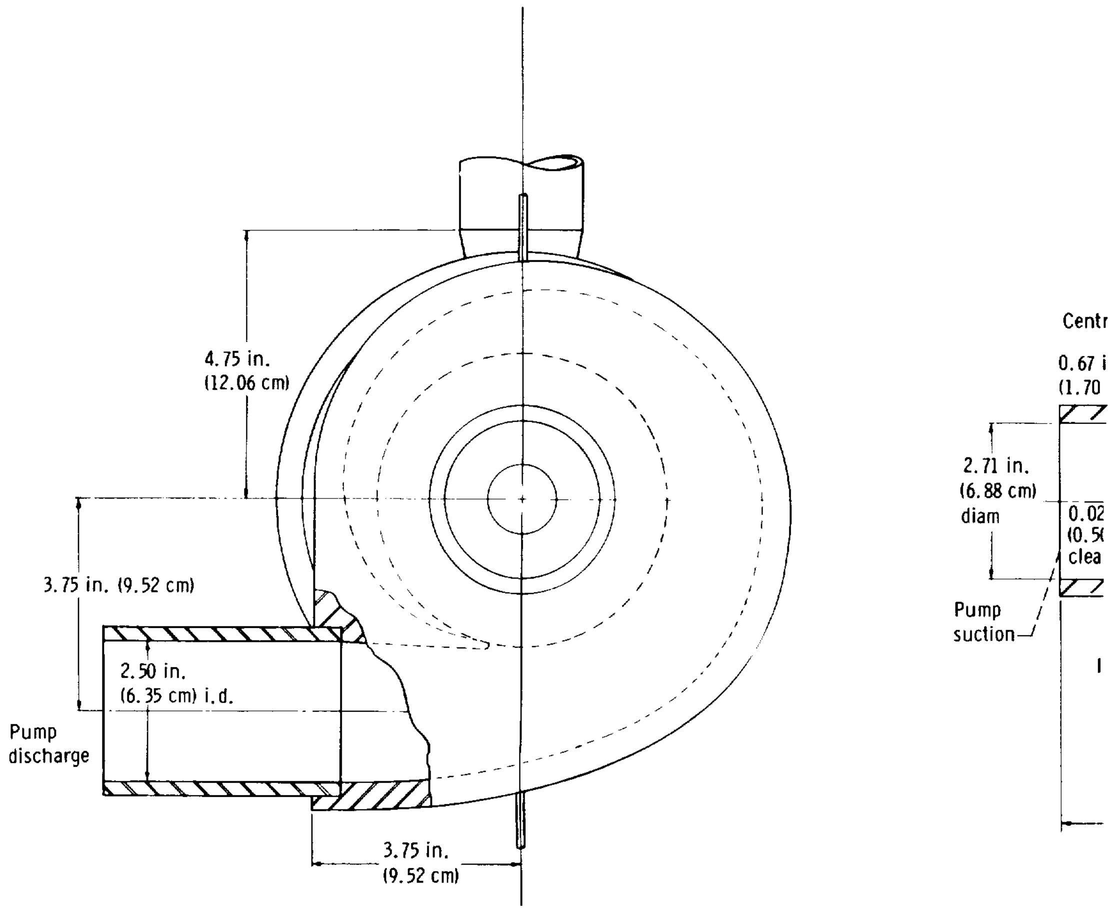


Figure 24. - Poison turbopump mechanical design layout. Moment of inertia of rotating assembly about axis, 0.085 inch pou (1560 cm^3); turbine weight, 70 pounds mass (31.8 kg); pump weight, 62 pounds mass (27.7 kg); pump wetted surface area, .

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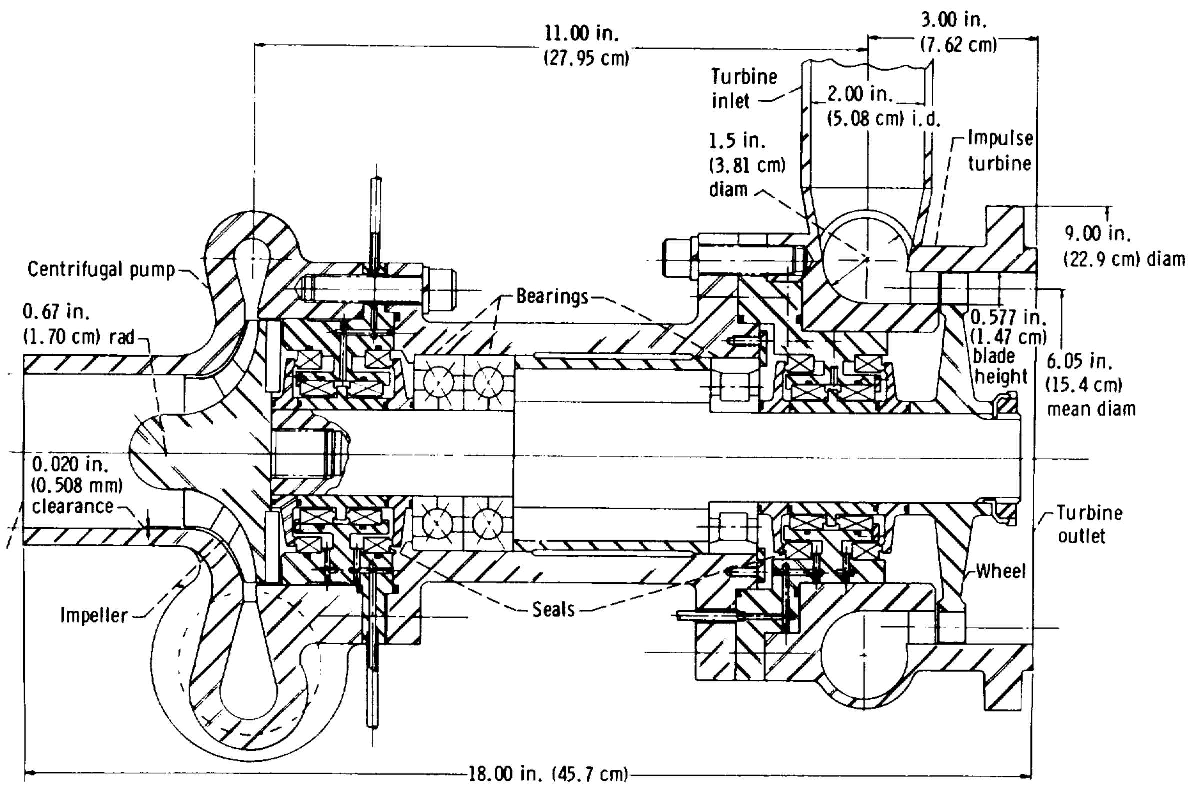
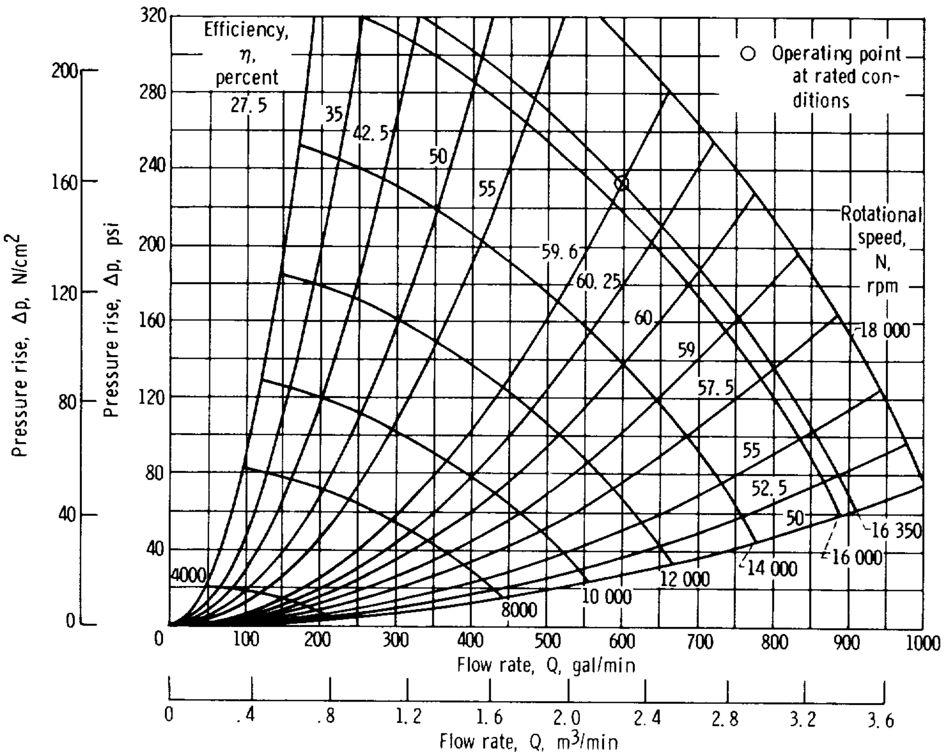
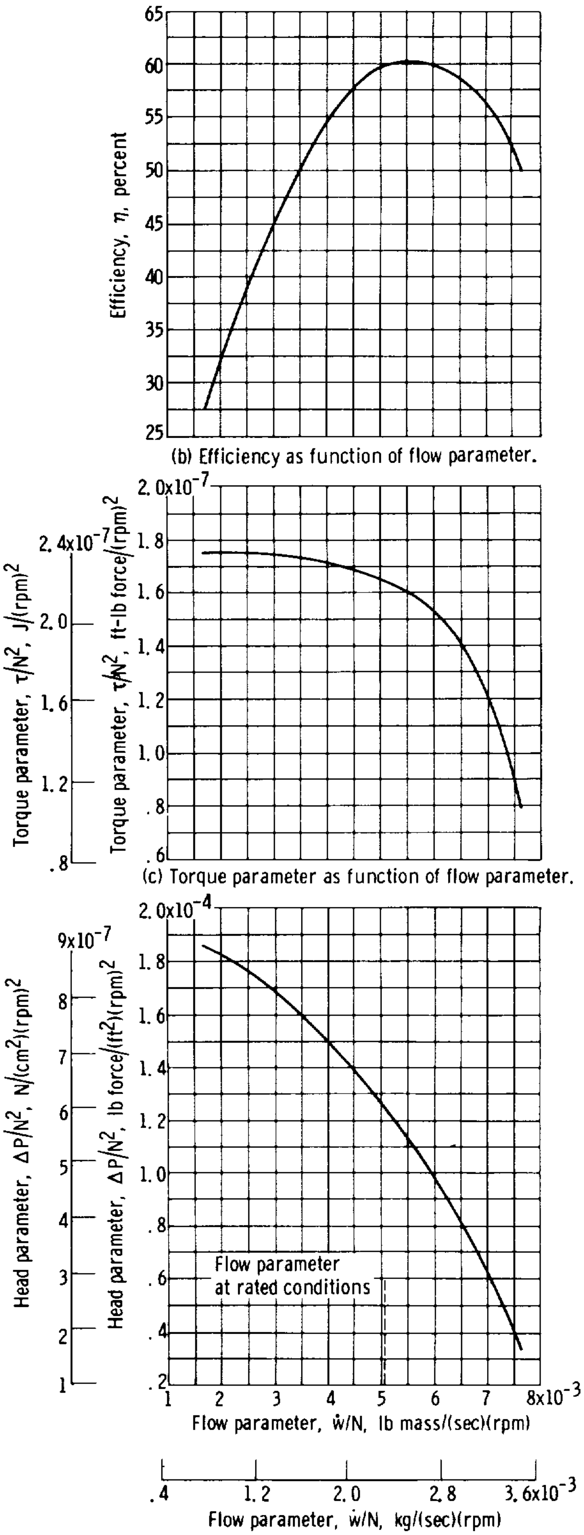


ch pound force second squared (0.00962 (kg)(m^2)); total rotating weight, 21 pounds mass (9.5 kg); turbine fluid volume, 95 cubic inches
area, 298 square inches (1920 cm^2); pump fluid volume, 69 cubic inches (1130 cm^3).



(a) Pressure rise as function of flow rate.

Figure 25. - Poison pump performance. Operating point at rated conditions: pressure rise, 233 psi (161 N/cm²); flow rate, 82.5 pounds mass per second (37.5 kg/sec) or 600 gallons per minute (2.265 m³/min); rotational speed, 16 350 rpm; water density, 61.7 pounds mass per cubic foot (988 kg/m³).



(b) Efficiency as function of flow parameter.

(c) Torque parameter as function of flow parameter.

(d) Head parameter as function of flow parameter.

Figure 25. - Concluded.

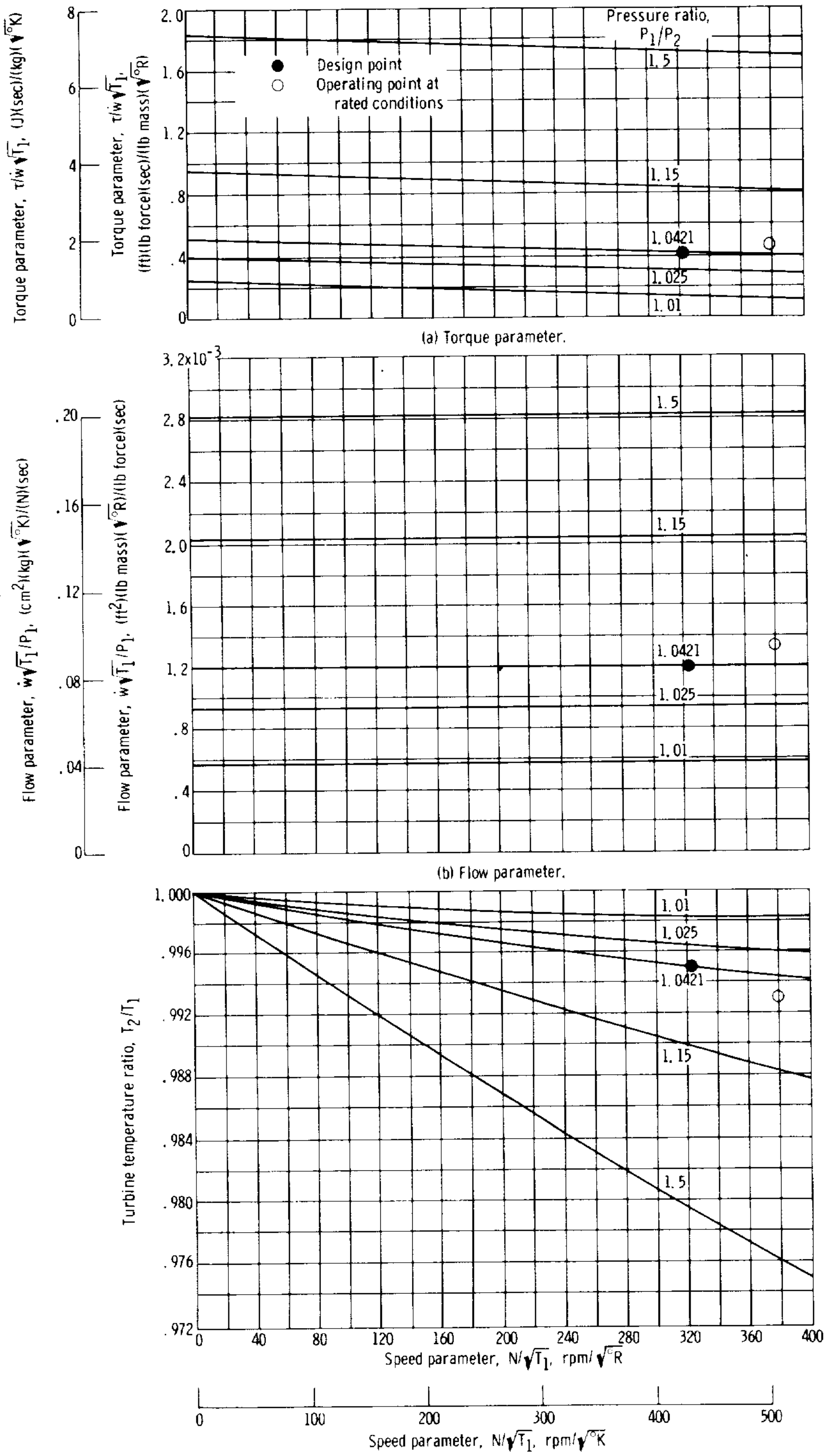


Figure 26. - Poison pump turbine performance as function of speed parameter. Design conditions: rotational speed, 13 940 rpm; power, 92 horsepower (68.6 kW); inlet temperature, 1860 °R (1030 °K); outlet temperature, 1851 °R (1028° K); inlet pressure, 480 psia (331 N/cm^2 abs); outlet pressure, 461 psia (318 N/cm^2 abs); flow rate, 1.927 pounds mass per second (0.875 kg/sec).